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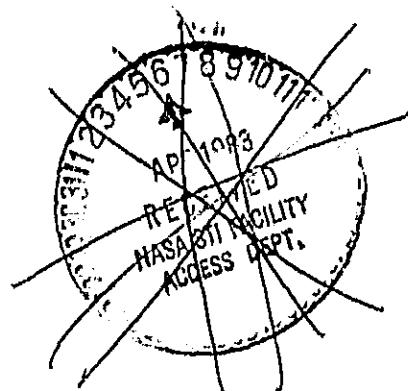
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INTEGRATED TECHNOLOGY ROTOR/FLIGHT RESEARCH ROTOR (ITR/FRR) CONCEPT DEFINITION STUDY

Raymond G. Carlson
Edward A. Beno
Harold D. Ullisnik



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RESEARCH ROTOR (ITR/FRR) CONCEPT
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FINAL REPORT

INTEGRATED TECHNOLOGY ROTOR/ FLIGHT RESEARCH ROTOR

CONCEPT DEFINITION STUDY

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Applied Technology Laboratory and Aeromechanics Laboratory
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PREFACE

The work reported herein was performed by Sikorsky Aircraft Division of United Technologies Corporation under Contract DAAK51-81-C-0030 for the Applied Technology Laboratory (ATL), U.S. Army Research and Technology Laboratories (AVRADCOM), Fort Eustis, Virginia, the Aeromechanics Laboratory (AVRADCOM), Moffett Field, California, and Ames Research Center, National Aeronautics and Space Administration, Moffett Field, California. The work was carried out under the technical cognizance of Robert Powell and Paul Mirick of ATL, Dr. Robert Ormiston and William Bousman of AL, and James Biggers of Ames Research Center, NASA. Sikorsky engineering personnel directly involved in the program include Dr. Raymond Carlson (Task Manager), Edward Beno, Harold Ulisnik and Gordon Miller.

The concept definition study is the concluding part of the first phase (Pre-design Studies) of the Integrated Technology Rotor/Flight Research Rotor (ITR/FRR) Project, a joint undertaking of the U.S. Army and NASA. Subsequent phases of the ITR/FRR Project are planned to provide for the design, construction, and flight test of advanced technology rotors. The concept definition study was undertaken to identify and evaluate potential rotor hub concepts, before initiation of the ITR/FRR preliminary design phase.



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ABSTRACT

Sikorsky Aircraft has completed the initial concept definition phase of the Integrated Technology Rotor/Flight Research Rotor (ITR/FRR) Program concluding that several designs offer the potential to meet the program goals. It was determined that varying degrees of success in meeting the overall goals can be achieved depending on which attributes of an advanced rotor design are considered most desirable. Several important aspects related to design attributes were identified as requiring better definition before the next phase of the ITR/FRR program commences.

Based on Sikorsky's previous experience with both bearingless and articulated rotor designs, plus recent developments in the use of composite and elastomeric materials, a number of advanced rotor system designs were initially conceived and investigated. From these, several were chosen that best meet the stated ITR goals with emphasis on stability, reduced weight and hub drag, simplicity, low head moment stiffness, and adequate strength and fatigue life. The study concluded that obtaining low hub moment stiffness was difficult when only the blade flexibility of bearingless rotor blades is considered, unacceptably low fatigue life being the primary problem. Achieving a moderate hub moment stiffness somewhat higher than state-of-the-art articulated rotors in production today is possible within the fatigue life constraint. Alternatively, low stiffness is possible when additional rotor elements, besides the blades themselves, provide part of the rotor flexibility.

Two primary designs evolved as best meeting the general ITR requirements that presently exist. An I-shaped flexbeam with an external torque tube can satisfy the general goals but would have either higher stiffness or reduced fatigue life. The elastic gimbal rotor can achieve a better combination of low stiffness and high fatigue life but would be a heavier design and possibly exhibit a higher risk of aeromechanical instability.

During the course of this study several ambiguities in the stated ITR goals were uncovered that should be clarified prior to the next phase of the program. In order to properly assess the fatigue life of a given design, a definitive rotor usage spectrum is required. Also the merit factor relating to the vulnerability requirement needs to be quantified. Finally, based on what are perceived as the most important goals, the weighting of each of the merit factors that define the overall rating of a given design should be revised.



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LIST OF SYMBOLS AND UNITS

A_v	Vulnerable Area, ft ²
b	Number of blades
E	Modulus of elasticity, lb/in. ²
EI	Bending stiffness, lb-in. ²
f	Rotor hub equivalent flat plate drag area, ft ²
H	Flexbeam height, in.
k_θ	Angular spring rate, in.-lb/rad
K_a	Merit factor probability that rotor system will be free from air/ground resonance instability
K_b	Merit factor equal to one half of the percentage by which the minimum rotor hub tilt angle exceeds the technical goal
K_c	Merit factor equal to qualitative estimate from 1 to 10, varying inversely with expected cost.
K_d	Merit factor equal to percent reduction from hub drag area technical goal
K_e	Merit factor equal to 5 if rotor hub moment stiffness is within $\pm 20\%$ of the technical goal; it is reduced from 5 by one-tenth of the percentage that the parameter exceeds a $\pm 20\%$ margin from the goal
K_f	Merit factor equal to ten times the probability of meeting or exceeding the fatigue life technical goal
K_m	Merit factor equal to one half of the percentage by which the parameter exceeds the minimum rotor hub moment technical goal
K_p	Merit factor equal to percent reduction from parts count technical goal
K_r	Merit factor equal to ten times the probability of meeting or exceeding the technical goal for MTBR



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LIST OF SYMBOLS AND UNITS (Continued)

K_s	Merit factor for torsional stiffness equal to -2 if pitch control system forces exceed 1.5 times typical pitch bearing hub value; equal to 0 if forces less than this level
K_v	Merit factor probability of surviving a small HEI projectile hit
K_w	Merit factor equal to percent reduction from hub weight technical goal
K_z	Merit factor equal to 0 to 2, qualitative estimate of practicality of incorporating auxiliary damping
K_{HM}	Hub moment stiffness, ft-lb/rad
M	Moment, ft-lb
$M_{E,Root}$	Blade root end ($r = 0$) edgewise moment, ft-lb
$M_{F,Root}$	Blade root end ($r = 0$) flatwise moment, ft-lb
N_p	Number of parts
r	Blade radial station measured from center of rotation, in.
R	Rotor radius, in.
t	Thickness, in.
W	Flexbeam width, in.
Z_T	Blade tip flatwise deflection, in.
α	Parameter defined by Reference 8 as function of ratio of shaft diameter to outer diameter of a circular disc
β	Hub angular rotation, deg
β_{TPP}	Tip path plane angle, deg
$\theta_{.75}$	Blade collective pitch angle at $r = .75R$
w_n	Blade natural frequency, rad/sec
Ω	Rotor angular speed, rad/sec



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1.0

INTRODUCTION

The Integrated Technology Rotor/Flight Research Rotor (ITR/FRR) Program has established a challenging set of specifications and design goals for a helicopter rotor system with the objective of obtaining a substantial advancement in the state of the art. The initial selection of a rotor hub configuration is a critical step if the design goals of the ITR/FRR are to be met. This study was undertaken to identify and evaluate potential rotor hub concepts before initiation of the preliminary design of the complete ITR/FRR rotor system. Numerous rotor hub concepts which could potentially meet the ITR/FRR requirements were examined. In addition to bearingless rotor concepts, which were required to be included in the study, other rotor hub configurations were considered. In particular, the elastic gimbal rotor emerged as a candidate for the ITR/FRR program.

The ability to provide sufficiently low hub moment stiffness (equivalent offset) and the means for providing blade pitch emerged as two key factors in the consideration of rotor hub concepts for the study. For bearingless rotors a combination of low flatwise flexibility and minimum hub size is needed in order to achieve low hub moment constants. Unfortunately, this leads to high stresses, static droop, and other potentially important considerations. The use of hub-to-shaft mounting flexibility in addition to blade flexibility can provide a means of obtaining lower hub moment constants without the complications caused by too low a value of blade flexibility. This approach led to some of the concepts considered in the study, including the elastic gimbal rotor. The blade pitch means is also an important consideration in defining a rotor hub. Bearingless rotors require, in general, a torque input to twist a flexible portion of the blade and thus provide blade pitch. Pitch shafts which roughly parallel the flexbeams or torque tubes which surround the flexbeams each have certain advantages. The selection of pitch means and the blade/hub flexibility approach are interdependent.

The study of rotor hub concepts emphasized the conceptual aspect of the problem. No detailed stress analysis was performed but enough was done to establish the approximate sizes of components as well as the appropriate material. This was necessary to establish the feasibility of the various concepts. Certainly, in the preliminary design phase which will follow, much remains to be done to detail the design in terms of layout, sizing, and stressing of the components.



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The results of this study should be considered for what they are intended to be: the study of rotor hub concepts which through engineering judgement, practical experience, and limited analysis are thought to be suitable candidates for the ITR/FRR program as presently understood.



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2.0

DESIGN SPECIFICATIONS AND GOALS

The rotor hub concepts considered in this study were selected and evaluated on the basis of their ability to meet the design specifications and goals of the ITR program and, more specifically, the rotor hub design specifications and goals. A ground rule adopted in the Sikorsky study was that the rotor hub concepts be suitable for use on a UH-60A helicopter. The rotor blade for the ITR has yet to be designed. It is recognized that subsequent design studies for an ITR blade to be integrated with the chosen hub concept may define a blade which has properties significantly different from the current UH-60A main rotor blades. However, for this study the blades were assumed to have the radius, chord, and tip speed of the UH-60A blades, with the blade weight distribution decreased by a constant percentage to satisfy the 7 $\frac{1}{2}$ of design gross weight goal for total rotor weight, as defined in the contract document. The design specifications and technical goals defined in the contract are given in the Appendix to this report.

The general specifications used by Sikorsky Aircraft in this study can be summarized as follows:

1. The rotor to have four blades.
2. The rotor to be compatible with the UH-60A. This was not a contract specification, but was considered as a practical requirement as the ITR rotor will ultimately be flown on an existing helicopter.
3. Design gross weight to be 16,000 lb. Note that the allowable gross weight range was 16,000 to 23,000 lb.
4. Rotor system to be free of aeroelastic and mechanical instability.
5. Rotor hub not to be incompatible with the incorporation of provisions for manual blade fold, surviving wire strikes, surviving combat damage from a small HEI projectile, and auxiliary lead-lag damping.
6. A minimum of one rotor hub concept to be bearingless.



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The technical goals used by Sikorsky Aircraft for the study include:

1. Rotor hub flat plate drag area to be not more than 2.8 ft².
2. Rotor hub weight to be not more than 400 pounds.
3. Rotor hub parts count to be less than 50, exclusive of standard fasteners.
4. A rotor hub moment stiffness of 100,000 ft-lb/rad was given as a goal. Other requirements indicate a value of 150,000 ft-lb/rad to be more reasonable. This higher value was used in the study.
5. The minimum conditions for no rotor hub fatigue damage to be 10,000 ft-lb vibratory moment and 5 degrees rotor disc angle.
6. Blade pitch control actuator force not to be substantially more than required by current rotor systems.
7. Rotor hub system fatigue life to be 10,000 hours.
8. Mean time between removal (MTBR) for the rotor hub to be 3,000 hours.
9. Manufacturing costs to be as low as possible without unduly compromising life cycle costs.
10. Rotor blade static droop to be comparable to current rotors.

With the exception of the hub moment stiffness and the static droop goal, the goals listed above are those given in the Appendix. It is probable that not all of the goals listed above can be met simultaneously. Design concepts were therefore considered on the basis of their ability to satisfy the goals to the maximum extent possible. Engineering judgement and experience, as well as use of the merit function, defined in the contract document and discussed later in the report, were used to make the trade-offs necessary in selecting rotor hub concepts and designs.



2.1

Design Loads

For this study, design loads for the ITR rotor hub have been based on those occurring on the UH-60A BLACK HAWK helicopter. This aircraft has a basic design gross weight of 16,825 lb. The ITR technical goals are specified with respect to a baseline aircraft having a gross weight of 16,000 lb. This 5% difference in design gross weight has conservatively been neglected by assuming the BLACK HAWK flight loads spectrum, which includes test data at gross weights up to 20,250 lb, for the ITR design loads. Measured BLACK HAWK flight loads are given in References 1 and 2.

The BLACK HAWK mission spectrum has been chosen to define the operating conditions for the ITR. This comprises hover, forward level flight, side flight, and maneuvering flight including turns, power dives and climbs, reversals, and pullouts to 3.0g. An example of the spectrum of flight conditions included is shown in Table I. Listed are the flight regimes that define the entire UH-60A usage spectrum including the percentage of time each occurs. The table also indicates the maximum measured flight test flapping angle at each condition. "High band" test data are tabulated and conservatively used for the ITR design loads. The "high band" data are the maximum measurements recorded during flight for a given condition. This is demonstrated by the flight test data shown in Figure 1. The degree of scatter shown, at a given level flight airspeed, is due to variables such as gross weight, center of gravity position, and altitude. From Table I the "high band" flapping during level flight ranges from ± 3.4 to ± 5.4 deg compared to angular motion as high as ± 14.3 deg during maneuvers. The present UH-60A rotor hub has a 10^8 cycles working endurance limit of ± 6.08 deg, thus resulting in no fatigue damage during level flight. Flapping angles indicated in Table I for maneuver conditions are the maximum values occurring during the transient condition. For certain regimes including extreme (evasive) maneuvers, GAG (ground-air-ground), reversals, pullouts, and droop stop pounding conditions, relatively high flapping angles are incurred. However these generally occur for only a small number of cycles compared to the percent time listed in Table I. The percent time shown corresponds to the length of the entire transient, not just the cycles at the maximum flapping angle shown. Actual fatigue life calculations take this factor into account by cycle counting at finite flapping levels within each transient.



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Figure 1 also indicates the UH-60A hub moment, calculated from the measured flapping angle, as a function of airspeed. The UH-60A has a hub moment stiffness of 175,000 ft lb/rad at normal operating rotor speed (258 rpm). This stiffness is defined as the moment, acting at the center of the hub, per unit angular rotation of the rotor disc about an axis perpendicular to the rotor shaft axis, the rotor disc being defined by the circle inscribed by hypothetical rigid blade tips.

A comparison of the UH-60A blade centrifugal force distribution and the corresponding assumed load for the ITR is shown in Figure 2. The ITR centrifugal force was derived as follows. An ITR technical goal is that the total rotor weight be 7.0 percent of the 16,000 lb design gross weight. An additional goal is that the rotor hub weight be 2.5 percent of the design gross weight. This implies that the blade weight goal would be 4.5 percent of design gross weight. Therefore, for a four-bladed ITR, the weight of one blade is 180 lb. A UH-60A blade weighs 210 lb. This gives a ratio of ITR to UH-60A blade weight of 0.857.

Assuming the ITR blade weight distribution is related to the UH-60A distribution by the constant .857 at any given radial station, the ITR curve shown in Figure 2 results. At the root end, the UH-60A centrifugal force is 70,000 lb and the corresponding ITR value is 60,000 lb. This value was used. It is possible that design requirements for the ITR blade might result in lower tip speeds and a lower centrifugal force, or that changes in the spanwise distribution of mass might increase or decrease centrifugal force from the value assumed.

The above described flapping, head moment, and centrifugal force loads were utilized along with other measured UH-60A flight loads to estimate the ITR design loads. Additional pertinent flight measurements include push rod load, lag damper load, and spindle bending moments.

3.0

SELECTION OF CONFIGURATIONS

Sikorsky Aircraft, in its ongoing research and development program for advanced rotor systems, has considered and evaluated a wide range of rotor hub concepts. From this, several have emerged as potential candidates for further development. This experience was drawn upon to narrow down the potential number of rotor hub concepts for evaluation against the ITR goals. In addition, rotor hub configurations discussed in the literature were considered for possible ITR suitability. From this review a number of candidate rotor hubs emerged. The rotor hubs considered are listed below.

A. Bearingless Rotors

1. I-beam flexbeam with torque tube.
2. C-beam rotor.
3. Anvil-strap rotor.

B. Rotors With Hub Flexibility

1. Elastic gimbal rotor.
 - a. Gimbal spring below blades.
 - b. Gimbal spring above blades.
2. Soft mounted rotor.

C. Articulated Composite Elastomeric Hub

The list includes bearingless rotors of several types. In addition, bearingless rotors with additional hub flexibility were considered. The additional flexibility helps to reduce the hub moment stiffness. The elastic gimbal rotor (EGR) is another candidate. It is a natural extension of the hub flexibility approach. The EGR uses a gimbal bearing with a hub spring to provide the majority of the hub moment flexibility. Also considered was a composite elastomeric rotor head. This rotor is a further development of the elastomeric bearing articulated rotor heads in service on current Sikorsky helicopters, including the S-76 and the UH-60A. This rotor hub though not bearingless still has many favorable attributes with regard to the ITR requirements.



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Each of the configurations in the list offers certain advantages. The composite elastomeric rotor head can satisfy a number of the ITR specifications and goals. However, two other concepts were selected as the primary concepts for this study. One is the bearingless rotor featuring a single I-beam flexbeam with an external torque tube. The second is the elastic gimbal rotor, which has a composite gimbal spring and an elastomeric gimbal bearing joining a bearingless-type rotor to the rotor shaft. Both of these rotors, as well as the composite elastomeric rotor, are discussed in the following sections, along with other significant rotor hub concepts which were considered in the study.

4.0

ARTICULATED COMPOSITE ELASTOMERIC HEAD

One concept considered for the ITR is an advanced articulated rotor constructed primarily of composite materials and utilizing elastomeric bearings to provide flap, lag, and pitch articulation. One possible version of this concept is shown in Figures 3 and 4. The primary member is a composite prelagged hub fitted and bolted to the main rotor shaft. It is comprised of graphite upper and lower plates bolted to a graphite main retention plate that is interleaved with fiberglass wraps for added fail-safety characteristics. The upper and lower plates provide the attachment points for lag dampers, rotating scissors, and optional bifilar vibration absorbers. The yoke is a fiberglass U-shaped structure consisting of unidirectional plies in the centrifugal direction with cross plies reinforcing the bolted attachment with the blade. The elastomeric bearing consists of inboard and outboard end plates, the yoke forming a strap to capture the inboard end plate while the outboard end plate engages the hub. In addition to capturing the elastomeric bearing in compression, the yoke reacts control inputs to the rotor via an attached pitch horn and internal torque box. This configuration has a high level of fail-safety. The elastomeric bearing's radial location allows for a prescribable hinge offset.

The composite elastomeric rotor head characteristics differ from existing articulated rotor heads in several important aspects. Increased fail-safety results from the redundancy inherent in the design configuration. Composite structures operating in fatigue at moderate strain levels have extremely long crack propagation times which allow a substantial interval between the point at which damage is first detectable and actual failure. This property allows an extra margin of safety since cracks are visibly detectable long before failure could ever occur. Although simple in design, the composite elastomeric design still requires lag dampers and flapping/droop stops. The well understood dynamics of this design results in a very high probability that the rotor would be free from instabilities throughout its operating envelope. Although vulnerability to catastrophic failure due to a small HEI projectile is considered improved compared to its metal counterpart, the concept results in a relatively high drag profile which necessitates fairing of the yoke. Rotor head weight is estimated to be 20 percent less than comparable metal articulated rotors and producibility requirements and service life improvements are expected to reduce life cycle costs.

5.0 BEARINGLESS ROTORS

5.1 Stiffness Requirements

The ITR goals and specifications have been used to guide the definition of the bearingless rotor designs considered in this Concept Definition Phase. In addition, previous Sikorsky research and development efforts under corporate and government funding (References 4, 5, and 6) provided preliminary direction in selecting geometry and materials with characteristics most likely to meet the ITR requirements. In particular, the unique properties of composite materials are expected to provide payoffs in cost, weight, and reliability and maintainability improvements.

The design of the flex member determines many of the characteristics of the bearingless rotor system. Important considerations in designing this member include required head moment stiffness, blade natural frequencies, allowable strain levels for long fatigue life, and ballistic and environmental tolerance.

In order to approximately size the flex member, the following criteria were used. Based on earlier Sikorsky bearingless main rotor studies, a minimum blade-to-flexbeam attachment point at 20 percent radius was assumed. The blade outboard of 20 percent radius is assumed to have the stiffness properties of the UH-60A main rotor blade, but its weight distribution was reduced to meet the ITR rotor weight goal of 7 percent of design gross weight. This resulted in a 14 percent reduction of the UH-60A blade weight outboard of 20 percent radius. A soft inplane rotor configuration with a blade inplane (edgewise) frequency in the range of .65 to .75 per rev was desired for stability and load considerations.

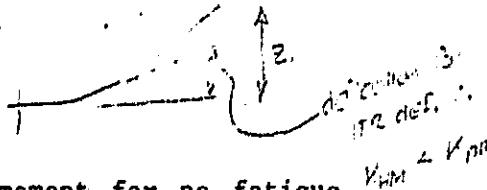
A design flatwise root end moment of $\pm 60,000$ in.-lb per blade was based on the ITR goal of a 10,000 ft-lb minimum rotor hub moment below which fatigue damage will not be incurred by the hub. In lieu of a specific ITR design goal for an edgewise root moment a value was determined based on existing soft inplane rotor designs. Design edgewise moments for existing hingeless rotors are indicated in Figure 5 as a function of gross weight. Extrapolating the existing soft inplane data to an ITR design gross weight of 16,000 lb results in an endurance edgewise root bending moment of $\pm 70,000$ in.-lb.

Initially both graphite epoxy and fiberglass epoxy flexbeams were considered. For each material, I-beams and rectangular-section beams were analyzed by a computer flexbeam optimization program. The major goal in using this analysis was to design a flexbeam which, when

joined to an outboard blade having the stiffness of the UH-60A main rotor blade, would result in an inplane frequency of .65 to .75 per rev. A secondary goal was to minimize the flexbeam cross-sectional geometry, and therefore weight, by achieving an approximately equal maximum allowable strain level along the entire flexbeam length. Lowering production costs was an indirect goal resulting from the minimization of the flexbeam geometry.

The optimization program was utilized to analyze flexbeams assumed to be fabricated entirely of unidirectional (radial) material. Accounted for in the analysis is a blade including a root end flexbeam joined to an outer scaled UH-60A main rotor blade. The program is a rotating blade analysis that calculates the first flatwise and edgewise modes. Subject to the design root end (center-of-rotation) flatwise and edgewise moments, the program calculates the response of each mode including all internal moments and shear forces acting along the entire blade. Centrifugal force effects are included for operation at normal (258 RPM) rotor speed. Torsion loadings are not included. Under the combined effects of the flatwise and edgewise modes, for a selected beam width the program determines the remaining beam geometry to meet the prescribed strain value. For the I-beam case, the flange and web thickness is determined for a selected beam height; for the rectangular-section beam the thickness is optimized. For the final flexbeam geometry the modal frequencies and flatwise and edgewise moment distributions are calculated by the program. Hub moment stiffness was determined from the response of the first flatwise mode. It is based on the modal moment at the root end of the flexbeam and the blade tip flatwise deflection, or algebraically,

$$K_{HM} = \frac{\frac{b}{2} M_{E, \text{Root}}}{Z_T \frac{R}{R}}$$



Meeting the 10,000 ft-lb minimum rotor hub moment for no fatigue damage was the controlling factor in using this analysis rather than trying to meet a specific hub moment stiffness goal.

Figures 6 and 7 indicate the results of the beam optimization program on the inplane (edgewise) frequency for graphite and fiberglass I-beams respectively. Shown are the effects of beam width, height, and thickness. Edgewise frequency is seen to be directly proportional to beam width and inversely proportional to beam height. At a given beam height, increasing the maximum flange and web thickness decreases the frequency, but the effects are not linear because the edgewise frequency is a function of both the width and thickness. In

the vicinity of a 0.7 per rev inplane frequency, the graphite I-beam's frequency is relatively sensitive to change in beam height while the fiberglass beam's frequency is almost unaffected by height. This is due to the higher modulus of elasticity of unidirectional graphite compared to fiberglass, resulting in smaller overall cross-sectional dimensions such that dimensional changes are more effective in producing stiffness change. A graphite I-beam having a width of 9 in. or greater can meet a 0.7 per rev inplane frequency goal. Below this width the beam section becomes a solid rectangle strained beyond the endurance limit before reaching the frequency goal of 0.7 per rev. A 9 in. wide by .85 in. high graphite I-beam with $t = 0.2$ in. will have the desired edgewise frequency of 0.7 per rev and an estimated head moment constant of 145,000 ft-lb/rad. A fiberglass I-beam having the same 0.7 per rev frequency requires a minimum 15 in. width (see Figure 7) with a height of 1.5 in. and flange thickness of .45 in. The estimated head moment stiffness for this fiberglass configuration is 185,000 ft-lb/rad. In each case the calculated head moment stiffness includes the centrifugal stiffening effect due to hub offset resulting from the effective termination of the inboard end of the flexbeam at a radial position equal to approximately one half the flexbeam width. Increasing the beam width while maintaining the same edgewise frequency, or increasing height with a given width, was found to increase the head moment stiffness for both graphite and fiberglass flexbeams.

During the course of determining the effects of the I-beam geometry on edgewise frequency and head moment stiffness, it was found that the shape of the graphite stress-cycles to fracture (S-N) curve conflicts with the specified goals of the ITR program. The ITR technical goals call for a minimum hub moment of 10,000 ft-lb and a minimum rotor tip path plane tilt angle of 5 deg, below which fatigue damage should not be incurred by the hub. An additional goal is a rotor hub system fatigue life of 10,000 hours. However, a fatigue load prorate computer program run with the UH-60A flapping spectrum (Table I) indicated that a flapping endurance limit of approximately 10 deg for graphite is required to meet a 10,000-hour fatigue life. For a given load spectrum and material, the prorate program calculates a curve of replacement time as a function of 10^8 cycles endurance limit. Due to the shape of the graphite S-N curve and the assumed UH-60A flapping spectrum, the resulting ITR endurance limit is 10 deg, twice the 5 deg ITR goal. For a hub moment stiffness of 150,000 ft-lb/rad and 10 deg hub angular rotation, the blade root end flatwise moment is:

$$M_{F,Root} = \frac{K_{HMB}}{b/2}$$

$$= \frac{150,000}{4/2} \left(\frac{10}{57.3} \right)$$

$$= 13,100 \text{ ft-lb}$$

$$= 157,000 \text{ in.-lb}$$

This is more than double the technical goal value. This points out the fact that the ITR goals of fatigue life, tilt angle, minimum no damage hub moment, and hub moment stiffness as specified are somewhat incompatible and difficult to meet without exceeding at least one of them. A given design then becomes driven by the most desirable or most stringent goal or goals. To determine the effect of raising the blade root end design flatwise moment from 60,000 in.-lb (ITR goal) to 157,000 in.-lb (10,000 hour fatigue life with UH-60A flapping spectrum) the flexbeam optimization computer program was rerun. The results are shown in Figure 8. Comparing Figures 6 and 8 indicates that for a chosen beam width the height and maximum flange thickness must be increased to meet the higher flapwise design moment. The minimum width necessary to achieve a 0.7 per rev edgewise frequency for the 157,000 in.-lb design moment case is 11 in. This configuration also results in the hub moment stiffness increasing to 205,000 ft-lb/rad, which is significantly higher than the ITR goal of 150,000 ft-lb/rad. For a constant design hub moment spectrum, lower hub tilt angles will result from the increased stiffness. Similar trends were found to exist for an I-beam meeting the fiberglass' 7.3 deg prorated flapping angle required to meet a 10,000 hour fatigue life.

What this means is that a possible inconsistency exists between the ITR goals of a minimum hub moment of 10,000 ft-lb for no fatigue damage and a 10,000-hour fatigue life. The key factor is the selection of a hub moment spectrum. Flight test data for the UH-60A BLACK HAWK define a hub moment spectrum which, if applied to the ITR, appears to necessitate a much higher minimum hub moment for zero fatigue damage if a 10,000-hour life is to be met. The moment spectrum to be used depends upon the intended aircraft mission. Therefore, a better definition is required, since the moment spectrum has a significant effect on the hub moment stiffness which is achievable.

In addition to determining geometry requirements for the I-shaped flexbeams, the beam optimization computer program was used to analyze rectangular section beams to the $\pm 60,000$ in.-lb flatwise and $\pm 70,000$ in.-lb edgewise design moment criteris. Figure 9 shows the results for rectangular graphite and fiberglass flexbeams. Meeting a 0.7 per rev edgewise frequency requires a 6-in.-wide graphite or an 8-in.-wide fiberglass beam. The thickness of the 6-in.-wide graphite beam tapers from .87 in. inboard to .82 in. outboard while the 8-in.-wide fiberglass beam thickness tapers from 1.8 to 1.1 in. Estimated head moment constant is 139,000 ft-lb/rad for the graphite design and 180,000 ft-lb/rad for the fiberglass configuration.

The I-beam configuration is considered preferable to the rectangular-section beam even though the minimum head moment stiffness is slightly higher (3 to 4%). Lower weight for the same stiffness is possible with the I-beam. Also, a rectangular-section beam may need to be longer due to expected higher torsional shear stresses.

As a final result of the bearingless rotor stiffness requirements study, the graphite flexbeam was selected as being superior in most respects to one made of fiberglass. Although material cost would probably be greater, the graphite flexbeam is lighter and its overall cross-sectional dimensions are smaller. It can also be optimized to produce a 20 percent lower head moment stiffness than a fiberglass beam.

5.2 Torque Tube/I-Beam Configuration

5.2.1 Description: Based on the ITR goals and specifications and the results of stiffness/geometry trade-off studies described previously, the primary ITR bearingless design evolved and is shown conceptually in Figure 10. The flexbeam component is a single, constant width, I-shaped cross section beam constructed of uniaxial graphite epoxy. It is compliant in terms of deforming as required due to outboard blade motions and control system inputs. Two stacked I-beams pass through and are clamped rigidly within BLACK HAWK tail rotor-type hub plates, each beam serving two opposing blades of the four-bladed rotor. The lower hub plate forms an integral part of the shaft adapter, thus permitting installation of the entire rotor head as an assembly. At approximately 20 percent rotor radius, the flexbeam is bolted to the outboard blade via an angled blade attachment fitting that precones the blade. The blade precone axis is chosen to coincide with the shear center axis of a composite torque tube which surrounds the beam. The torque tube provides a torsion path from the pitch control rod to the outboard end of the torsionally soft I-beam. By increasing the torque tube wall thickness below the flexbeam, its



shear center axis can be prescribed to lie along the preconed blade axis as shown in Figure 10, thus minimizing unfavorable twist-bend couplings. The lower torque tube wall terminates in a bolted pitch arm fitting that attaches to the pitch control rod. A sliding pin extension from this fitting passes through the centering ball of a snubber assembly which acts to restrain the motions of the inboard end of the torque tube. A snubber housing extension of the central hub piece provides a rigid enclosure for the snubber assembly and a droop stop mechanism. Due to the relatively low flatwise stiffness of the flexbeam required to meet the ITR head moment stiffness goal, static droop of the blade is due to one-g gravity load is in excess of an acceptable limit unless a droop stop is used to stiffen the system at low rotor speeds. The droop stop uses centrifugal force to swing the stop out of the way as rotor speed increases. As the rotor slows, a spring returns the droop stop to a position providing a solid restraint between the pin extension from the torque tube and a hub-mounted support. It is possible than an up-flap stop may be required for start-stop operation in high ambient wind conditions, but it has been assumed that it is not needed. In order to decrease rotor head drag, a fairing is included surrounding the hub plates and snubber housings as shown in Figure 10. Blade folding for this concept could be provided at the torque tube to outboard blade joint.

Meeting the ITR goals for head moment stiffness (120,000 to 180,000 ft-lb/rad) and minimum hub moment for no fatigue damage (10,000 ft-lb) is the critical design driver of a torque tube/I-beam bearingless rotor. Based on the parametric stiffness studies discussed in the previous section, a 9-in.-wide by .85-in.-high I-shaped flexbeam with flange and web thickness of .20 in. inboard tapering to .11 in. outboard has been chosen for the soft inplane design. Blade properties outboard of the torque tube are scaled from the UH-60A blade to meet the ITR rotor system weight goal of 7 percent of design gross weight (1120 lb). These properties were input to a coupled natural frequency computer program, the results of which are shown in Figure 11 at $\theta_{75} = 0$ deg. The predominant motion in each of the coupled modes is shown in the figure by F (flatwise), E (edgewise), and T (torsion). As indicated, the predominant motion can be a function of rotor speed. At normal rotor speed of 258 RPM the .73 per rev inplane frequency is within the desired range for a soft inplane rotor for freedom from ground and air resonance instabilities. A head moment stiffness of 175,000 ft-lb/rad results from the first flatwise natural frequency at 1.04 per rev which is relatively low for a bearingless rotor and corresponds to an equivalent rotor offset of 5.7 percent radius. This head moment stiffness value is somewhat higher than the value calculated using the beam optimization program. The remaining frequencies shown in Figure 11 are considered to be



only approximations, since design of the outboard blade was beyond the scope of this study. Actual blade design would include ways to detune any modes that are too close to multiples of rotor speed in order to minimize the possibility of blade response amplification. An example is the flatwise bending mode (3rd bending) at 2.94 per rev which would cause unacceptable 3 per rev flatwise response. By tailoring the outboard blade weight and/or stiffness distribution, this modal frequency can be moved sufficiently away from 3 per rev to reduce the amplification of blade response due to 3 per rev forcing frequency.

The snubber assembly was sized to restrain the inplane and out-of-plane motions of the torque tube at its inboard end. An estimate of the inplane motion was made using a coupled natural frequency computer program that models the flexbeam, torque tube, and outboard blade properties. By running the program without modeling the snubber, the relative motion between the hub and the torque tube is determined. Assuming a vibratory lead-lag motion of ± 2 deg of the blade tip, the resulting inplane motion at the inboard end of the torque tube is $\pm .50$ in. The ± 2 deg lag motion is based on maximum measured motion of existing articulated rotors. Based on typical operating strain in existing full scale elastomeric tail rotor snubbers, a vibratory shear strain of ± 36 percent was conservatively specified. The snubber was also designed to a flatwise (axial) load of ± 1500 lb, again conservatively based on UH-60A measured push rod loads and the shear load at the snubber as predicted by the natural frequency program at the design condition. Based on the above design requirements and the properties of elastomers used in similar applications, the snubber configuration shown in Figure 10 was defined. Shear pads of laminated elastomer, 3 by 3 by 1.5 in. thick, meet the design strain limits. The elastomeric snubber shown will also act to provide structural lag damping to the blade. Additional damping material can be added within the snubber housing if this is found necessary to solve a stability problem that might arise during the ITR development phase.

Overall, the torque tube/I-beam bearingless rotor design offers improvements over existing rotor state-of-the-art designs in many of the merit function factors of the ITR specifications. Its major attribute is low weight. Its simple design leads to a low parts count while the utilization of light weight composite materials with long crack propagation times leads to improved R&M characteristics. Lag dampers are not required although the concept offers the ability to add additional damping in the form of an elastomeric component in

the snubber housing. The risk of aeroelastic instability is considered low based upon general experience with soft inplane non-matched stiffness flexbeam design. The cumulative effect of the low weight simple design with improved R&M characteristics should be reduced manufacturing costs in addition to low life cycle costs. Although a blade folding capability is possible at the torque tube to blade attachment joint, it would add additional complexity to the design and has not been included in the layout shown in Figure 10.

5.2.2 FRR Modifications: Several properties of the torque tube/I-beam bearingless rotor design can be modified to provide a range of configurations for Flight Research Rotor (FRR) testing. These are described below.

1. Flexbeams - Hub moment stiffness could be varied by changing the flexbeams. Change of material or section geometry would offer the ability to determine effects on blade natural frequencies, aircraft handling qualities, vibration, and blade/flexbeam stresses.
2. Structural Damping - Augmented structural damping can be accommodated in the hub to evaluate its effect on rotor dynamics or to solve a potential stability problem.
3. Torque Tube - The torque tube's shear center can be varied by changes in material or section geometry, and its effect on dynamic coupling evaluated.
4. Pitch Horn - The pitch horn location can be varied to change the pitch to flexbeam coupling. This can be accomplished by designing new pitch horns and push rods and reorienting the swashplate relative to the hub.
5. Sweep, Droop - These hub angular offsets can be varied by modification of the flexbeam to blade attachment fitting.
6. Rotor Blade - Blades of differing planform, twist, section properties, and advanced tip geometries can easily replace the baseline blade design.

5.3 Soft Mounted Rotor

Although the torque tube/I-beam bearingless rotor design described above meets most of the primary ITR goals, a variation of this concept was developed that has the capability to exceed some of these requirements. However, while certain merit factors can be improved, others are degraded. The soft mounted rotor is described here as an optional design that could be desirable if more emphasis is placed on certain ITR design requirements. A conceptual drawing is included as Figure 12.

The primary advantage of the soft mounted rotor is its inherent capability to provide a lower head moment stiffness than other bearingless designs. Designs that clamp the flexbeams rigidly between stiff hub plates are incapable of achieving an effective hinge offset below approximately 5 percent radius. By utilizing no-maintenance elastomeric hub shear springs, the soft mounted rotor can reduce the effective offset below this limit, possibly to as low as 3 percent radius. The hub springs can be designed to provide a selected head moment stiffness by prescribing the geometry and composition of the elastomer. As shown in Figure 12, the hub springs are mounted between split arms of the flexbeams and within the rigid hub structure, thereby reacting, through shear deformation, the bending motions of the flexbeams. The back-to-back elastomeric springs on each blade's flexbeam are preloaded in compression against each other and transmit the rotor driving torque from the hub plates to the blades. The connection of the hub springs to the upper and lower hub plates is through an I-beam shaped member shown in the side view of the hub. This member also serves as the attachment for the snubber. Head moments are almost entirely reacted by these springs since, at the center of rotation, the flexbeams possess a teetering boundary condition in the flapping direction due to the use of rotationally soft elastomeric trunnion bearings. Besides acting to center the flexbeams in the blade radial direction, the trunnion bearings react rotor thrust. The split flexbeams and location of the centering bearings will allow the mounting of optional mast-mounted equipment. Splitting the flexbeam as shown in Figure 12 also eliminates the high stress concentration factor associated with a single flexbeam with an elliptical hole in the center.

Outboard of the elastomeric hub springs, the concept is similar to the torque tube/I-beam configuration. An aerodynamically shaped torque tube surrounds the flexbeam and transmits control inputs to the outboard blade. A snubber, connected to the hub through the hub

spring mounting, will act to restrain inplane and out-of-plane motions of the torque tube's inboard end. It is positioned between, but not connected to, the split arms of the flexbeam. A bolted joint would form the connection between the outboard end of the torque tube, the flexbeam, and the blade.

Low offset and the resulting low head moment stiffness are the main attributes of the soft mounted rotor. This rotor concept offers a means of achieving lower head moment stiffness and possibly high rotor head fatigue life and tip path plane tilt endurance limit. Actual fatigue life and tip path plane endurance limit will be dependent on the desired head moment stiffness, since this stiffness controls the required elastomeric hub spring geometry. Conversely, the space available for these hub springs is limited by other considerations such as hub drag and flexbeam geometry.

Along with the noted advantages of the soft mounted rotor compared to the torque tube/I-beam rotor, certain disadvantages exist. The risk of aeromechanical instability is greater due primarily to its untested state and the possibility of greater dynamic coupling resulting from the teetering concept. Also, hub weight and parts count are higher due to the elastomeric hub springs and trunnion bearings. Finally, manufacturing costs would be higher, again due to the added elastomeric elements.

5.4 C-Beam Rotor

Another bearingless rotor which was considered is the C-beam rotor. It was originally considered in the study because it offers great simplicity in its design. Its primary disadvantage was the difficulty in designing the C-beam rotor to provide a sufficiently low hub moment stiffness. In addition, a droop stop needed for such a configuration would be difficult to include. Therefore, the C-beam was initially dropped as a candidate. Later in the study, however, it was reconsidered in the arrangement shown in Figure 13. The reason for its reconsideration was again its simplicity. The single C-beam has a shear center which is behind the beam and on the blade pitch axis, which is also the quarter chord of the outboard blade. As the shear center is behind the beam, the pitch shaft can be conveniently located on the shear center. This reduces coupling between blade deflections and control motions. Shear forces at the pitch shaft attachment will not produce twisting of the C-beam, and conversely torsional moments at the outer end of the C-beam will not deflect the attachment point. The attachment can be moved off the shear center to provide coupling if desired. The configuration shown in Figure 13 uses a pitch shaft which is structurally integral with a

trailing edge fairing. This arrangement increases the torsional stiffness, permitting the use of a smaller diameter tube. It should also improve the vulnerability characteristics of the pitch shaft. An increase in the edgewise stiffness of the pitch shaft also occurs due to the structurally integral trailing edge fairing. A radially flexible attachment at the outer end of the pitch shaft as well as a spherical bearing at the inboard end are used to reduce inplane bending moments in the shaft and thus minimize the effect of the higher edgewise stiffness in the pitch shaft. The torque tube connection to the blade is intended to transmit torque only. This is accomplished by floating (axially) the inboard end of the torque tube in a self-aligning dry bearing and by providing a "soft" flatwise and chordwise connection between the torque tube and blade at the outboard end.

The flatwise stiffness of the C-beam used in this configuration is greater than that required to meet the desired hub moment stiffness. The equivalent hinge point would be 9 to 10 percent, thereby obviating the need for droop stops. The C-beam transition to the outboard blade structure occurs in a region where the steady centrifugal moment tends to be cancelled by the steady drag moment, making the transition section not a critically loaded area for normal operation. In autorotation, however, this cancellation does not occur. Further analysis is required to assess the structural risk. Composites would be used for fabrication of the C-beam and blade structure.

The development of this configuration has not been extensive enough to judge fully its potential suitability. If low hub moment stiffness is a critical requirement, this configuration would not be suitable as shown. It could be combined with hub flexibility in some form to lower the hub moment stiffness. If a requirement for low hub moment stiffness is not critical, this configuration could be attractive.

5.5 Anvil-Strap Rotor

Another bearingless rotor hub concept investigated was the anvil-strap rotor shown in Figure 14. This configuration was pursued as a possible means of reducing the effective offset or hub moment stiffness by using two flexible straps guided in flatwise deflection by curved surfaces at the outboard end of the hub plates. Preliminary analysis indicated that the desired hub moment stiffness could not be achieved without significant further development in the design and fabrication of the flex straps and their attachment. Analysis indicated that the stress concentration factor associated with the pin wrapped composite-elastomeric straps is a potential problem.



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Additionally, the efficiency of the wrapped joint is a factor, since relatively thick straps are required to meet a static design requirement of 4 times centrifugal force. In addition, the design is complicated by the need for droop and anti-flap restraint. Therefore the rotor hub did not warrant further consideration as a preferred configuration.

6.0 ELASTIC GIMBAL ROTOR

6.1 Rotor Hub Concept

The elastic gimbal rotor (EGR) concept is basically a bearingless rotor attached to the rotor shaft through a gimbal. A spring with selected stiffness across the gimbal provides the desired control moment characteristics between rotor and airframe and serves as part of the rotor head fairing to minimize hub drag. The spring also transfers rotor torque from shaft to blades. The use of a gimbal greatly reduces 1-per-rev blade flapping and thus reduces inplane Coriolis forces and the resulting inplane blade vibratory moments. The flatwise stiffness of the inboard region of the blade is not critical to the hub moment stiffness; blade flatwise stiffness can be greater than that of an equivalent bearingless rotor. This simplifies the design requirements for the rotor blade. Figure 15 shows the concept as developed for the ITR program. Design criteria are based in part on an aeroelastic study of the EGR (See Reference 7).

The heart of the idea is the ability of the rotor head to tilt relative to the shaft with elastic restraint to provide a constant speed joint without mechanical complexity or 2-per-rev vibration associated with conventional mechanical universal joints. The hub tilt degree of freedom greatly reduces blade flapping relative to the hub for any given amount of tip path plane inclination relative to the shaft. This suppression of flapping motion minimizes the Coriolis effects which result from blade motion relative to the plane of constant rotational speed. Thus, the elastic gimbal rotor is expected to have some fundamental stress and vibration advantages over conventional bearingless rotors. This could permit a system weight savings. Another potential virtue is that the EGR rotor blades may be made relatively stiff in flatwise bending, avoiding critical stress concentration situations while still having a low rotor equivalent flapping hinge offset which reduces high speed gust response and fundamental vibration excitation levels. The stiff blades also permit rotor startup or shutdown operations in high wind conditions without encountering excessive blade motion excursions or requiring the complexity of a blade flapping lock system at low rotational speeds.

The EGR for the ITR is, as its name implies, a fully gimballed rotor employing elastic gimbal restraint to allow the rotor to produce hub moments. The EGR is gimballed on a single elastomeric bearing similar to those used in several state-of-the-art rotor systems. The use of an elastomeric bearing to provide the complete function of a universal joint provides both an elegantly simple load path as well

as a gimbal whose failure mode is benign (i.e., it is fail operative and easily inspected). This elastomeric gimbal bearing is preloaded against a dry self-lubricating spherical bearing whose effective center is coincident with the elastomeric bearing and whose function is to carry negative g loadings occasionally encountered in flight. The preload is set such that negative g loads will not fully unload the elastomeric gimbal bearing, thereby assuring that the bearing is never loaded in tension. The central titanium hub piece provides the connection between the blade assemblies and the elastomeric gimbal bearing. Blade centrifugal forces are transmitted through the central hub to be cancelled out by the diametrically opposite blade. The lift force is carried directly by the hub tail shaft, through the elastomeric gimbal bearing, into the main rotor shaft. A central hole in this hub piece provides for access to rotor mounted equipment such as sights or antennas. Figure 15 also shows a shaft adapter that is used to interface the EGR with either the UH-60A BLACK HAWK main rotor shaft, or in a slightly different version, the RSRA's S-61 main rotor shaft.

Head moments for aircraft control are generated in the EGR by providing a restraining spring in the form of a graphite filament wound diaphragm spring, looking much like an upside down "Frisbee", which is attached to the main rotor shaft. Its outer rim attaches to the blade flexbeams via a series of titanium clamping fittings. A dual element spring with one spring nested inside the other provides the stiffness required at acceptable stress levels. The gimbal spring not only provides the desired hub spring restraint but is also the direct path by which main rotor shaft torque is transmitted from the main shaft to the blades. The EGR's low aerodynamic drag and anticipated low radar cross section are the direct result of the shape of this gimbal spring and the covering dome-shaped Nomex®-honeycomb/fiberglass fairing.

The rotor blade flexbeams of fiberglass composite construction outboard of the gimbal spring attachment points become two open back-to-back I-beams. These beams transmit all blade loads directly to the central hub while simultaneously allowing for blade pitch change, some flapping, and edgewise restraint. Blade pitch control is input via the graphite torque tubes, grounded on the inboard end in self-aligning bearings and attached at their outboard ends to the blades themselves. Pitch control is input to these torque tubes via the forged aluminum horns. Torsionally flexible fairings enclosing

the flexbeam torque tube members provide for a low aerodynamic drag interface between the EGR hub and the rotor blades themselves. Blade folding, if required, could be accommodated by a joint connecting the outboard end of the flexbeams with the blade proper, in a manner similar to folding arrangements available on the other bearingless rotor configurations.

Another arrangement of the EGR (Figure 16) illustrates a second means by which blade fold can be provided. In this arrangement the blade flexbeams originate at fold fittings just outboard of the gimbal spring element. The pitch torque tube section in the hub is supported at its outboard end by the hub blade fold fitting and provided with one half of a disconnect coupling. The portion of the torque tube in the blade is supported at its inboard end in the blade root end fold fitting and provides the other half of the disconnect coupling. By removing either one of the split cone pins (similar to those used on BLACK HAWK) the blade can be folded around the remaining pin.

The elastic gimbal hub concept lends itself to the achievement of low hub weight, low-cost fabrication, low hub drag, and good aerodynamics in the blade root area, factors vital to any high speed helicopter application.

6.2 Inplane Damping

Blade inplane damping for an EGR could be obtained by adding a constrained layer of an elastomeric damping material along the flexbeams. This is shown schematically in Figure 17. A thin strip of material, either metal or a composite, bridges the two flexbeams on the top and bottom surface. This coverplate is flexible flatwise and in torsion, but is stiff edgewise. It is bonded to the flexbeams through a damped elastomeric interface. The amount of damping can be regulated by choice of damping materials, by the length of the cover plate and by the bonded area and thickness of the elastomer. By bonding the elastomer to the flexbeams over a limited area at each end the effect on blade stiffness would be relatively small. The structure for auxiliary damping would logically be integrated with the torsionally flexible fairing enclosing the flexbeams.

Damping is provided by relative motion between the flexbeams and the coverplate. Edgewise motion of the blade inflight consists largely of inplane bending of the flexbeams with rigid body motion of the blade proper (outboard of the flexbeams). The flexbeam inplane curvature due to bending can be used to provide damping, if the cover plate is stiff enough so that it will not bend inplane. There are at

least two approaches which can be used to obtain damping from the differential motion as shown in Figure 17. If the same elastomeric material is used at the inner and outer end, damping occurs for relative rotational motion of the flexbeam and the coverplate at both ends. There is some damping due to radial shear as well. If the outer end of the coverplate is rigidly attached to the flexbeam and an elastomeric material used at the inner end only, damping is produced primarily by inplane shear across the inboard elastomeric damper.

6.3 Component Sizing

Critical components in the elastic gimbal rotor are the gimbal spring, flexbeams, pitch shaft, and the gimbal bearing. The physical characteristics of these components were determined by a combination of analysis and the use of test data from similar components. These components are discussed below. The remaining components were sized based upon either simplified analysis or comparison with comparable UH-60A components.

6.3.1 Gimbal Spring: The gimbal spring size was scaled from 1/5-scale model test data for a gimbal spring which was a dual element spring made of fiberglass. This spring was tested for stiffness and fatigue. Fatigue tests were at amplitudes up to 9 degrees for about 10×10^6 cycles with no structural failure or stiffness deterioration. Therefore, it is believed that a gimbal spring scaled from this configuration will be successful. For scaling, a formula from Reference 8 was used. This defines the thickness to stiffness relationship of a constant thickness circular disc restrained at the outer radius with a moment applied to a central shaft. The formula is

$$t^3 = \frac{\alpha k_\theta}{E}$$

where t is thickness, k_θ is the angular spring rate, E is the material modulus of elasticity, and α is a parameter defined in Reference 8 as a function of shaft diameter to outer diameter of the circular disc. This formula when applied to the model gimbal spring test data, gave a thickness value corresponding to the thickness at 35 percent radius. See Figure 18. The formula was then applied to the ITR gimbal spring to define a thickness at 35 percent, and the thickness distribution was scaled using the model thickness distribution. This gave the values in Table II for fiberglass and graphite

gimbal springs. A graphite spring was chosen for the EGR rotor, as it was lighter. For this calculation, a gimbal spring stiffness of 195,000 ft-lb/rad was used. Since a part of the total hub moment stiffness depends on blade flexbeam flexibility, which is a spring in series with the gimbal spring, the net hub moment stiffness for the combination is approximately 139,000 ft-lb/rad., less than the stiffness of the gimbal spring alone.

6.3.2 Flexbeams: The flexbeams are made of fiberglass wrapped at ± 15 degrees from the radial axis. The beams were sized, in addition to strength, to provide the required rotor natural frequencies and hub moment stiffness in combination with the gimbal spring. Preliminary sizing was obtained by scaling flexbeams previously designed for another rotor. Comparison of these scaled values with design requirements showed them to be acceptable. Flatwise EI was 16.2×10^6 lb-in. 2 compared with 5.0×10^6 lb-in. 2 for the bearingless configuration. A higher flatwise stiffness can be used since the gimbal spring rate is the dominant term for the hub moment stiffness. This fact simplifies the flexbeam design requirement. An edgewise EI value of 92.9×10^6 lb-in. 2 for the pair of flexbeams compares with 240×10^6 lb-in. 2 for the bearingless rotor. This lower value is required by an EGR, as the root end of the flexbeam is at a larger radius than in a bearingless rotor. The flexbeam area is sufficient to withstand centrifugal force greater than 5 times the design centrifugal force, thus exceeding the usual 4 times centrifugal force requirement.

Blade natural frequencies are shown in Figure 19. The first edgewise natural frequency is at 0.7 per rev at normal rotor speed. The first flatwise natural frequency is at 1.11 per rev for the collective mode (blade cantilevered at the hub) and at 1.03 for the cyclic mode (blade pinned at the centerline). The placement of the blade natural frequencies, especially the higher modes, depends upon the properties of the outboard blade. For the purposes of this study they were scaled from UH-60A properties in order to meet the general design goals for the ITR. Further definition of the blade properties and rotor speed is needed and constitutes part of the next phase of the ITR program.

6.3.3 Pitch Shaft: The pitch shaft is made of graphite tape wound at 45 degrees to the tube axis. This provides maximum torsional stiffness with the lowest bending stiffness. Figure 20 shows a plot of the effect of pitch shaft diameter on the torsional natural frequency of the blade, assuming UH-60A control system flexibility and scaled UH-60A blade properties. An O.D. of 2.9 inches was selected to keep the torsional natural frequency above 4 per rev.



6.3.4

Gimbal Bearing: The gimbal bearing was sized using approximate formulae for strain allowables and fatigue life. The size of the bearing is similar to the elastomeric flap-lag bearing used in the UH-60A although the loads on the gimbal bearing are smaller. Thus, the formulae can be used with confidence, since their validity has been demonstrated by UH-60A experience. Gimbal bearing life is greater than 3,000 hours for a hub tilt angle of 5.1 degrees (equivalent to about 8 degrees tip path plane tilt) at a sustained load factor of 3.5. When prorated by a suitable flapping and load factor spectrum this should produce a fatigue life in excess of the 10,000 hour requirement. More detailed analysis will be required once the spectrum is established.

6.4

Manufacturing Considerations

The elastic gimbal rotor has a number of features which can potentially lead to low manufacturing costs, not the least of which is its simplicity and small number of parts. The metallic components present no particular difficulty in manufacture. The gimbal bearing is a state-of-the-art elastomeric bearing, similar to bearings presently in service. It is anticipated that the gimbal spring element will be made by winding resin impregnated graphite fiber tape on a mandrel similar to the procedure used in making model specimens. The assembly is then cured with externally applied pressure. Automation of the winding process can make this a low cost process. The pitch shaft will also be made of graphite fiber, wound and cured on a tube to provide a stiff lightweight shaft at low cost. The flexbeams are fiberglass. Each flexbeam is made by filament winding over a shaped mandrel to give a rectangular section of high torsional stiffness. Prior to curing, the flexbeam is post-formed to an I-beam shape in the central sections of the beam. This gradually transitions to a rectangular section at both ends. This post-forming technique has been demonstrated successfully. It provides a low-cost means of making flexbeams having the desired flatwise and edgewise flexibility.

6.5

FRR Modification

The EGR rotor provides the potential for an extensive number of rotor hub variations which could be made to a Flight Research Rotor (FRR). These are listed below.

1. Gimbal Spring - The rotor hub moment stiffness could be varied by changing the gimbal spring. Material, section thickness, or geometry could all be changed.

2. Blade Natural Frequencies - By replacing the flexbeams, a wide range of blade modal frequencies could be obtained. Stiff inplane blades as well as soft inplane blades could be tested. Since hub moment flexibility is controlled primarily by the gimbal spring, blade inplane stiffness can be varied with less concern for effects on flatwise flexibility. This would not be true for a conventional bearingless rotor. Figure 17 illustrates one manner in which edgewise damping could be added should that prove to be desirable for the soft inplane investigation.
3. Torsional Natural Frequency - The pitch shaft stiffness can be varied by changing material, winding angle, or size to vary torsional natural frequencies.
4. Prelag, Sweep, Droop, Precone - These effects can be varied by changing the attachment of the flexbeam outer ends to the blade proper or by modifying the hub plate.
5. Pitch Horn - The pitch horn location can be varied to change pitch to gimbal coupling. This would be done by designing new pitch horns and push rods and by reorienting the swashplate relative to the rotor hub.
6. Rotor Blade - The blades themselves could be replaced with blades of differing planform, twist distribution, section properties, or any number of advanced tip caps.
7. Blade Fairing - The blade fairing covering the flexbeam can be replaced by alternative configurations.

6.6

Alternate Configuration for an EGR

Another configuration considered in the study was an elastic gimbal rotor in which the gimbal spring is mounted above the rotor system. Such an arrangement is shown in Figure 21. As compared with the preferred EGR configuration, this alternate configuration had higher drag, was more complex, and was heavier. The gimbal spring now required separate mounting hardware; the rotor torque path became longer, enlarging the shaft and thus the gimbal bearing required; the rotor fairing was enlarged and became a separate structure. The configuration was therefore not selected for continued study.



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7.0

RSRA INTEGRATION

Any of the proposed ITR rotors can be readily installed on the RSRA with relatively little modification of RSRA components and few new parts. These items can be summarized as follows:

1. New swashplate
2. New rotating push rods
3. New rotating scissor links
4. Raised position of RSRA swashplate ball
5. Modified planetary stage of transmission
6. Modifications to blade severance system

The modification of the RSRA to accept ITR rotors requires a minimum of new parts because the ITR rotors have been conceived for installation on the UH-60A BLACK HAWK. The present BLACK HAWK rotor system employs an intermediate shaft extension designed to allow the rotor head to be lowered for air transportability. The use of slightly different shaft adapters permits installation of the ITR rotor on either the BLACK HAWK or the S-61 main gearbox of the RSRA.

The new swashplate assembly, push rods, and scissor links are required to match the travel ranges of the RSRA control system with the travel ranges required by the ITR rotors. In addition, the ITR rotor plane of rotation is raised six inches above the current RSRA's rotor plane of rotation in order to provide adequate blade root to rotor pylon fairing clearance.

The modifications required to the blade severance system are fundamentally to adapt the system from a five-bladed system to a four-bladed system. These changes result in significant reduction in system complexity particularly in the rotary transfer unit that controls the sequencing and direction of rotor blade jettison from the aircraft.

The main gearbox of the RSRA takes the 18,966 RPM inputs from two T-58 engines and reduces this to 203 RPM at the main rotor shaft. In order to provide the assumed higher RPM of the ITR rotors (258 RPM if UH-60A tip speed is assumed) the last reduction stage (the planetary system) is redesigned to obtain a 3.6468 to 1 ratio versus its current 4.6296 to 1 reduction ratio. Since the planetary is the last



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stage of gearing in the main transmission, all the other gears remain unchanged and carry the same power and torques as they do now. Thus the accessories and tail drive systems remain unaffected. Note that should a different RPM be selected for the ITR, the gear ratio would be changed accordingly.

No modifications of any kind are required to the tail rotor system of the RSRA as a result of changing the main rotor shaft's output RPM since the reduction ratio between the engine inputs and the tail rotor drive shaft output is not affected if a change is made to the main gearbox's planetary reduction stage. The tail rotor RPM, therefore, remains unchanged from its current value and no modifications to the tail rotor system, as a result of installing the ITR on the RSRA, are anticipated. It may be necessary however to change the tail rotor control rigging values as a result of the differing main rotor torque values between an ITR rotor and the current S-61 rotor on the RSRA.

The RPM change is required to match the requirements for the assumed smaller diameter ITR rotor as compared to the current RSRA rotor. The ITR rotor will then have somewhat less hover capability than the current RSRA aircraft now enjoys (due to the somewhat higher disc loadings), but analysis has shown (Reference 9) that more than adequate flight time is available to an ITR-equipped RSRA despite the slightly reduced fuel load that would be carried.

Other considerations such as airframe vibration levels, control system load limits, main rotor shaft bending load limits, or any of a number of stability considerations that could potentially arise from the substitution of a four-bladed system for a five-bladed system will receive further design attention in subsequent phases of the ITR work. However, preliminary analyses accomplished so far indicate that none of these considerations should result in any significant truncation in the RSRA's current flight envelope.



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8.0

VULNERABILITY

Vulnerability is that measure of the likelihood of a system or component being disenabled when exposed to a threat environment. Various levels of disenabling or "kills", such as mission abort, forced landing, aircraft attrition and so forth can be defined and measured. For the ITR study the threat environment assumed has been a small HEI projectile and the kill level assumed is complete attrition. A rotor component is vulnerable, if having sustained a hit the resulting failure would result in the immediate loss of the helicopter of which the rotor is a part.

A good relative measure of vulnerability is vulnerable area. Vulnerable area of a component is expressed as the presented area of that component in a plane normal to the shot aspect of the threat, multiplied by the probability of a kill of that component given that a hit has occurred. To obtain this, the first step is to establish a complete description of the rotor concept and identify the flight critical components. Damage analyses are then conducted on the individual components to establish which ones, if impacted by a small HEI round, would fail in such a manner as to result in complete loss of the helicopter. For the ITR study we used multiple shot lines from two mutually orthogonal shot aspects: straight up from underneath, and straight in from the side. On some shot lines a component may be masked by another. For example, a hub piece might mask a flexbeam. Some components such as dual straps may be redundant on one shot line but singly vulnerable on another shot line. For those components shown vulnerable in the damage analyses, the vulnerable areas for the various shot lines are then summed for each shot aspect to produce a total vulnerable area for each shot aspect. The vulnerable area totals for each shot aspect are then averaged to establish a mean vulnerable area for the entire rotor system.

The vulnerable areas have been computed for two views only; from underneath and from the side. The rotors are considered to be isolated in space so that no reduction in vulnerable area results from the rotor being masked from underneath by the presence of a fuselage. This simplifying assumption plus the averaging of only two views instead of six, results in the computation of a larger numerical value of vulnerable area than is actually the case.

Table III includes the vulnerable areas of the ITR rotors considered during this phase of our work. In computing these areas, all rotor hubs were assumed to extend from their centerline of rotation out to their 32-inch radial station. Note that rotors using flexbeams are typically more vulnerable outboard of the hub than articulated



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typically more vulnerable outboard of the hub than articulated rotors. This is due to the more sensitive stability boundaries and generally higher state of strain at which they operate.

In addition, the rotors were treated as isolated objects. Thus the presence of other objects such as the helicopter itself which could result in producing a fragmented or tumbled round impact on some shot lines was not considered. Therefore, the vulnerable areas presented in Table III should not be considered absolute vulnerable areas but rather as measures of relative vulnerability between the various rotor concepts evaluated.

9.0 MERIT FUNCTION

The contract called for evaluation of the selected rotor hub configurations using a defined merit function:

$$\text{Merit Function} = K_v \times K_a \times (K_d + K_w + K_p + K_e + K_m + K_b + K_r + K_c + K_f + K_z + K_s)$$

The two multipliers are K_v , the vulnerability merit factor, and K_a , the aeromechanical merit factor. These and the remaining merit factors which are summed are defined in accordance with the contract definition in the following section.

9.1 Merit Factor Definition

9.1.1 Vulnerability: The probability of surviving a hit is defined as the probability of a component continuing to function after having sustained a defined damage level resulting from a specified threat. For the ITR specified threat of a small HEI projectile, none of the rotor hubs studied can survive all possible shots; they all possess some vulnerable area to some shot lines from some threat aspects.

Vulnerable area, the presented area of the rotor hub in a plane normal to the shot line that cannot sustain a hit and continue to function, is considered a better measure of vulnerability. These areas have been determined for the rotor concepts studied and are shown in Table III. Simplistically, the smaller the number, the higher the probability of survival becomes for a given threat intensity. However, if hit in their vulnerable components, all of the rotors could fail resulting in a K_v of zero if K_v were to be computed as the probability of surviving any possible hit. Therefore, a vulnerability merit factor, K_v , was calculated for each rotor using

$$K_v = \left(1 - \frac{A_v}{16}\right)$$

where A_v is the vulnerable area in square feet, and 16 ft^2 is a reference area. The reference area was chosen to be clearly larger than the actual value for any current hub. Selection of a larger reference area would make the vulnerability factor less sensitive to differences in vulnerable area.

9.1.2 Aeromechanical Stability: The merit factor for aeromechanical stability was estimated based upon relative risk of a significant developmental cost to achieve aeromechanical stability. Zero risk is assigned a factor of 1.0. Bearingless rotor stability with soft



inplane blades has been demonstrated. However, some risk of instability exists for any specific bearingless configuration in combination with a given airframe. Therefore, a risk factor must apply to any bearingless rotor. Stability requirements for an elastic gimbal rotor have been calculated in an analytical study, and gimbal rotors have flown. The EGR has yet to be confirmed by test. Based upon this lack of test confirmation, a lower merit factor was assigned to the EGR than to bearingless rotors.

9.1.3 Drag: Drag was determined using existing Sikorsky methodology. This methodology includes the use of the Generalized Rotor Performance (GRP) program and a semi-empirical approach (Reference 10) for determining the incremental hub drag. A trending equation based on hub swept frontal area is used to determine the drag of the hub in free flow. Interference effects due to hub/pylon and hub/shaft are included as incremental drag coefficients added to the hub drag coefficient. The interference drag is then adjusted to account for the strength of the flow disturbance generated by the hub or shaft and the general flow stability on the aft pylon.

Ellipsoidal faired hubs are treated somewhat differently. The base drag of the fairing is determined as a function of the crest point pressure of the fairing. This is a function of the thickness to diameter ratio of the fairing. Incremental coefficients for skin friction, exposed blade shank interference and the shaft are then added to the base drag coefficient to determine the total hub drag.

In order to obtain a direct drag comparison between hub configurations the hub was defined as extending from the center of rotation to the UH-60A hub blade attachment point. Therefore any torque tube sections inboard of the UH-60A blade attachment location were included as part of the hub. This attachment occurs at 10% of blade radius (32 in.). These drag increments were determined by modeling the parts as airfoils and then computing the equivalent parasite drag area increase caused by the change, using the Sikorsky Generalized Rotor Performance (GRP) program. In this way increased torque requirements are included in addition to the flat plate drag increment associated with the appropriate pitch angles.

The Generalized Rotor Performance Analysis is Sikorsky's standard method for predicting main rotor forward flight performance. This analysis calculates rigid blade flapping and lead-lag response through timewise integration of the blade equations of motion. Blade torsional response is not modeled. Aerodynamic inflow can be modeled as constant or harmonic input from a variable inflow forward flight wake analysis. The analysis uses tabulated airfoil data and treats

stall, compressibility and yawed flow effects in the calculation of rotor performance. When used to establish performance trends in preliminary design studies, GRP is run with a uniform inflow determined from momentum considerations. When detailed results are desired, the analysis is performed using variable inflow obtained through automatic iteration with the Prescribed Wake Inflow Analysis.

In computing drag areas for the various concepts only the hub and blade attachment (i.e. torque tube) contributions were included. Rotor shaft and interference effects were neglected, since these depend upon the airframe as well and can be greatly affected by the airframe geometry. See Section 9.2.3 for drag values. The drag area, f , calculated for each hub concept was used in the merit factor calculation.

The drag merit factor K_d becomes

$$K_d = \left(\frac{2.8 - f}{2.8} \right) \times 100$$

9.1.4 Weight: Rotor hub weight was established by directly calculating component volumes from the two-view drawings and multiplying these by the appropriate densities of the materials involved. The resultant detail weight breakdowns were then summed to establish total rotor hub weight. The rotor hub was assumed to end at the 32-inch radial station (10% of rotor radius). The flight control push rods, rotating scissors, and shaft adapters were not included in the estimates.

The weight merit factor K_w becomes

$$K_w = \left(\frac{400 - Wt}{400} \right) \times 100$$

9.1.5 Parts Count: Parts count was established by simply counting parts. However, defining what is a part makes the task a little more sophisticated. To evaluate the parts count merit factor in this study, a part was defined as a piece, or component, of the hub that can be removed as an entity unto itself and after removal cannot be disassembled further. For example: an elastomeric bearing is counted as a single part even though it has an inner metallic member, multiple elastomeric elements and shims, and an outer metallic member. Similarly, a blade pitch control torque tube with the pitch horn bonded to it is counted as a single part despite the fact that in its fabrication it is produced from several pieces.

K_p , the parts count merit factor, is calculated as

$$K_p = \frac{(50 - N_p)}{50} \times 100$$

Standard fasteners are not counted, but a unique "fastener element" is (i.e., a special nut, washer, or safetying element).

9.1.6 Rotor Hub Moment Stiffness: This value was calculated for each configuration based upon the modal characteristics of the first flatwise mode. Root end bending moment was divided by the angle defined as $\tan^{-1}(Z_T/R)$, where Z_T is the tip deflection. Hub moment stiffness is this moment times $b/2$, where b is the number of blades. For the blade stiffness characteristics used, the hub moment stiffness is defined by flexbeam stiffness and hub flexibility alone, with blade stiffness having a negligible effect near the hub moment stiffness goal. The goal was defined in the contract as 100,000 ft-lb/rad, but other considerations led to the use of 150,000 ft-lb/rad as the goal.

The merit factor K_e equals 5 if the rotor hub moment stiffness is within ± 20 percent of the goal. It is reduced from 5 by one-tenth of the percentage that the parameter exceeds a ± 20 percent margin from the goal. Thus,

$$K_e = 5 - 10 \left[\frac{K_{HM} - 180,000}{150,000} \right] \quad \text{if, } K_{HM} > 180,000$$

or

$$K_e = 5 - 10 \left[\frac{120,000 - K_{HM}}{150,000} \right] \quad \text{if, } K_{HM} < 120,000$$

9.1.7 Fatigue Life, Minimum Rotor Hub Moment, and Minimum Rotor Hub Tilt Angle: The goal for minimum rotor hub moment for no fatigue damage was 10,000 ft-lb. The merit factor, K_m , equals one half of the percentage by which the parameter exceeds the technical goal.

$$K_m = 50 \left[\frac{M - 10,000}{10,000} \right]$$

The goal for minimum rotor hub tilt angle for no fatigue damage is 5 degrees. This angle was interpreted as the tip path plane angle resulting from hub flexibility and flexbeam flexibility. The merit factor, K_b , equals one half of the percentage by which the parameter exceeds 5 degrees.

$$K_b = 50 \left[\frac{\beta_{TPP} - 5}{5} \right]$$

The goal for rotor system fatigue life was 10,000 hours. The merit factor, K_f , equals 10 times the probability of meeting or exceeding 10,000 hours.

9.1.8 Reliability: Reliability was estimated based upon comparison of the rotor hub concepts with experience for the UH-60A rotor hub. The merit factor, K_r , equals 10 times the probability of meeting or exceeding the goal of 3,000 hours mean time between removal (MTBR).

9.1.9 Manufacturing Cost: The relative merit factors for cost were based upon estimates made by Sikorsky engineers. The merit factor, K_c , can vary from 1 to 10 and varies inversely with expected cost. Estimates were based upon the number of parts and their complexity, the weight of the system and upon special cost advantages of particular hubs.

9.1.10 Auxiliary Lead-Lag Damping: The merit factor, K_d , can vary from 0 to 2, depending on the feasibility of adding auxiliary lag damping if it should prove necessary in practice. All concepts considered can incorporate a practical means of lag damping.

9.1.11 Torsional Stiffness: The merit factor, K_s , has a value of -2 if the pitch control system forces exceed 1.5 times typical pitch control forces for current rotor systems. $K_s = 0$ if the forces do not exceed this value. The means of varying blade pitch was selected to minimize coupling forces for each configuration. In addition, flexbeam torsional stiffness was kept low to minimize basic twisting moment requirements. Therefore, all configurations have the potential to meet the torsional stiffness requirements so that $K_s = 0$ for all.

9.2 Merit Functions for Selected Rotor Hubs

The merit functions for the elastic gimbal rotor, the I-beam/torque tube bearingless rotor, the articulated composite-elastomeric hub, and the production UH-60A hub (included for comparison) are summarized in Table IV. The elastic gimbal rotor has a higher score (76.7) than the I-beam bearingless rotor concept. Note that if the I-beam is stiffened to meet fatigue life requirements at the cost of hub moment stiffness, it scores slightly higher (85.9) than the EGR. The stiffened I-beam results in a head moment stiffness exceeding the goal, but more pertinent to the scoring of the merit function as defined, the hub moment value for no fatigue damage is more than doubled. The resulting value for K_m , the hub moment merit factor, is

80.8, more than two-thirds of the total score. Thus, the merit function value for this bearingless rotor is overly influenced by a single term. The I-beam bearingless rotor which meets the hub moment stiffness requirement scores only 11.3, only slightly above the UH-60A production hub score of 3.6. The articulated composite hub had the highest merit function score when the vulnerability and stability multipliers were included. Note, however, that the high score resulted primarily from the moment and tip path plane angle merit factors. Even so the good score is indicative of the favorable attributes of this configuration.

The high EGR merit function was due to generally good scores in all factors with particularly high values in the drag and parts count merit factors. The I-beam bearingless rotors merit factors covered a wide range, the stiffer beam having a much higher score for the hub moment and fatigue life factors, while not being penalized significantly for missing the hub moment stiffness goal. Both of the articulated hubs had widely ranging merit factors. The production hub scored well in drag and hub moment, but poorly in weight and parts count. The composite hub scored well in the fatigue-related terms but poorly in drag, weight, and parts count.

The merit factors shown in Table IV can have fairly large values for those factors for which there is no limit on the allowable range. These factors are drag, weight, parts count, minimum hub moment, and minimum hub tilt angle. The high values tend to distort the relative merit functions of the rotor hubs. In particular, merit factor values for minimum hub moment or tilt angle can be quite large. However, these quantities are not significant in themselves. They relate to fatigue life for which the merit factor has a limit value of 10. It was shown that sizing a flexbeam for a larger minimum hub moment value would pay off in a substantial increase in the hub moment merit factor at the price of a relatively small penalty in the hub moment stiffness merit factor. This did not seem to be within the spirit of meeting the technical goals and specifications of the ITR.

For this reason, a second merit function was calculated. This is shown in Table V. The merit factors in Table V have been scaled from the values in Table IV. The scale factors were chosen to reflect their estimated relative importance. The multipliers are shown in the third column. Hub moment stiffness factor was doubled. The drag and weight merit factors were reduced to 20% and 60%, respectively, of their original values. This represents their relative importance based upon tradeoff studies used in the original UH-60A design studies. The parts count merit factors were reduced to 15% of their

original values. Minimum hub moment and tilt angle for no fatigue damage were reduced to 10 percent of original values, since they pertain primarily to fatigue life, which has a separate merit factor. Others were unchanged.

The resulting modified functions score the EGR highest, but the relative differences are much smaller. The EGR score is 26.2. The I-beam bearingless rotor hub scores slightly higher (19.3) than the articulated composite hub (14.8). The stiffened bearingless rotor hub shows a much smaller effect of the stiffness increase, but it is still enough to place it second behind the EGR. The UH-60A production hub scores lowest, penalized largely by weight and parts count. The revised merit functions give a better comparison between hub concepts, as no single terms dominate the score to the extent they did in Table IV.

A comparison of the various merit factors for the various rotor hubs shows their relative advantages.

9.2.1 Vulnerability: All hubs had similar vulnerable areas and differed little in their score (Table III lists the vulnerable area as well as drag, weight, parts count and hub moment stiffness used.)

9.2.2 Aeromechanical Stability: The articulated rotor hubs scored 1.0, as they do provide aeromechanical stability. The bearingless rotor configuration was given an 0.85 value based upon the potential development required to achieve stability in combination with a selected airframe. The EGR has yet to be demonstrated in test, and on that basis was given the lowest score.

9.2.3 Drag: The EGR had the best drag factor score, followed by the UH-60A production hub. Hub drag was assumed to be the drag of the hub alone. An equivalent drag area was added to account for the increased torque requirements due to the inboard torque tube drag on the bearingless configuration. Rotor shaft, body and fairing interference, and vibration absorber drag were not included. For example, the basic UH-60A hub drag is 2.0 ft^2 . Inclusion of the bifilar absorbers raises the drag value to 2.6 ft^2 . The further addition of the rotor shaft and interference effects increases the drag area to 5.4 ft^2 . Only the basic hub drag of 2.0 ft^2 was used. The drag areas were summarized in Table III. They are further defined in Tables VI and VII. Table VI gives projected area and the equivalent drag coefficient, which when multiplied by the projected area gives the hub drag. Adding torque tube drag then gives the total hub drag area used. The drag of the basic bearingless rotor hub is very low, but the torque tube effects increase the drag area significantly.

Table VII provides a compilation of component swept areas and drag coefficients used in the determination of the EGR and I-beam hub drag values. For the EGR, the blade shank area was included in the drag analysis. However, since the shank is small and elliptical in shape the drag increment is small (See Table VII). The I-beam hub has a smaller diameter fairing but includes inboard torque tube sections represented by thick double ended or conventional airfoils. The drag increment of the I-beam torque tubes was determined using the Generalized Rotor Performance (GRP) program in order to include increased torque requirements. This is accomplished by computing an equivalent parasite drag area from the drag and increased torque requirements.

9.2.4 Weight: The lightest rotor was the I-beam rotor. The EGR rotor was 8% above the 400-lb goal. The production UH-60A hub was the heaviest, but the composite hub provides a 20% weight reduction in an articulated rotor hub.

9.2.5 Parts Count: The EGR has the fewest parts. The bearingless rotor parts count was higher because of the need for droop stops. The articulated composite hub also required droop and flap stops, and it scores low. The production UH-60A hub scored by far the lowest in the factor.

9.2.6 Hub Moment Stiffness: All concepts satisfied the goal except for the stiffened I-beam. The penalty imposed for exceeding the goal was relatively small. Because of the impact of hub moment stiffness on high speed stability, vibration, and gust response, this penalty might in fact be too small.

9.2.7 Minimum Hub Moment: The articulated rotor hubs and the stiffened I-beam were designed to much higher no-fatigue-damage moments in order to meet fatigue requirements. Therefore, they scored very well. The EGR meets the requirement with a small expected margin. The gimbal spring is the critical elements in this rotor. The blades can be stiffened sufficiently to provide good fatigue life.

9.2.8 Minimum Tip Path Plane Tilt Angle: The articulated hubs can exceed the minimum tilt angle for no fatigue damage by good margins and score very well. The EGR distributes the deflection between gimbal spring and flexbeam. This enables it to exceed the goal of 5 degrees. The I-beam bearingless rotor hub does not meet the 5 degree requirement. The 10,000-ft-lb limit at the higher hub moment stiffness value of 175,000 ft-lb/rad is reached at 3.3 degrees.

9.2.9 Reliability: Reliability was estimated based upon expected life of components and ease of inspection. The UH-60A hub has less than a 3,000-hour MTBR, and therefore has a zero probability of meeting the requirement. The bearingless rotor and the articulated composite head could be designed to meet the requirement and are given the highest score. Difficulties in predicting full-scale gimbal spring characteristics and the location of an elastomeric bearing within the hub structure lowered the EGR score.

9.2.10 Manufacturing Cost: Cost factors were estimated based on complexity, weight, and parts count. All should be lower cost than the present production hub. The EGR gets a slight advantage based on simplicity of the components used and the relative weight and parts count.

9.2.11 Fatigue Life: Fatigue life of 10,000 hours is not met with the production hub and would be difficult to meet with the bearingless rotor. Both score low. The EGR does not have the problem of low blade flexibility. Therefore, the key to its meeting the 10,000 hours life is the gimbal spring and gimbal bearing. A 50 percent probability was assumed pending further work on the components.

9.2.12 Auxiliary Lead-lag Damping: All systems can be adapted for damping, the articulated hubs most easily. They were given a score of 2.0. The others have a score of 1.5.

9.2.13 Torsional Stiffness: The hubs were designed with low torsional stiffness and with pitch inputs along the outboard blade quarter chord to minimize control loads. They were given the nominal score of 0.

10.0

CONCLUSIONS AND RECOMMENDATIONS

This study has examined a number of rotor hub concepts and defined two which are suitable for the ITR/FRR program. These are the elastic gimbal rotor and a bearingless rotor using an I-beam flexbeam and a torque tube. Several others emerged as potential candidates, depending upon detailed requirements. These include the soft mounted rotor (a bearingless rotor with a flexible hub) and possibly the C-beam rotor, if a stiffer configuration is acceptable.

The study pointed out the difficulty in obtaining low hub moment stiffness using only the flexibility of bearingless rotor blades. Adequate fatigue life is the principal problem, although the need for droop or flap stops also poses problems. The fatigue life question is a difficult one to evaluate. It requires a better definition of the intended rotor usage spectrum than is available in this study.

There are several directions in which the design concepts could move to avoid the problem of having to provide both the required hub moment stiffness and adequate fatigue life. The past approach was to allow significantly higher hub moment stiffness, resulting in relatively simple hubs but with excessive vibration, gust response and angle of attack stability at high speeds. Alternately, low hub moment stiffness can be obtained by providing hub flexibility and not relying on blade flexibility alone. The blade stiffness could then be increased to provide adequate fatigue life. Both the EGR and the soft mounted rotor use this approach. The EGR has been examined in some detail and has been shown to provide a number of beneficial characteristics, including the potential for lower vibratory forces. The soft mounted rotor concept was developed toward the end of the study and has not been evaluated as fully, but it too offers potential benefits.

A further extension of the hub flexibility approach is the articulated rotor. The composite hub elastomeric bearing articulated rotor remains an excellent choice, especially for near-term usage.

In summary, certain conclusions can be drawn:

1. A bearingless rotor featuring an I-beam flexbeam and a torque tube can satisfy the general ITR/FRR requirements, but would have either somewhat higher hub moment stiffness or reduced fatigue life.
2. The elastic gimbal rotor is a good candidate for the ITR/FRR, satisfying the general requirements and goals.



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3. The use of hub flexibility in addition to blade flexibility improves the potential for meeting ITR/FRR goals.
4. The rotor hub concepts chosen can be flown on an RSRA with relatively limited aircraft modification. Definition of the complete ITR/FRR rotor design is needed to define fully the changes required.

Several recommendations for continuation of the ITR/FRR program also resulted from the study.

1. A better definition of the intended rotor usage spectrum is needed in order to define rotor fatigue life.
2. The vulnerability requirement needs to be quantified. A vulnerable area goal is suggested.
3. More realistic merit function factor limitations should be defined based on the relative importance placed on each goal.

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TABLE I. UH-60® MAIN ROTOR FLAPPING SPECTRUM

No.	Regime	Low - Mid G.W. (Up to 17,000 Lb.)			High G.W. (17,001 to 21,000 Lb.)			Any G.W. (Up to 21,000 Lb.)		
		Occur per 100 Hours	% Time	Max. Vibr. Flapping Flight Load ± Deg	Occur per 100 Hours	% Time	Max. Vibr. Flapping Flight Load ± Deg	Occur per 100 Hours	% Time	Max. Vibr. Flapping Flight Load ± Deg
1	Control Check									
2	Rotor Engagement									
3	Taxi									
4	Taxi Turns									
5	Rotor Shutdown									
6	Hover									
7	Left Side Flight	1.411	4.17	0.200	5.27	0.050	3.88	28	375	3.76
8	Right Side Flight	0.200	6.71	0.200	6.71	0.050	5.24	29	.049	12.95/5.38
9	Rearward Flight	0.200	4.61	0.200	4.61	0.050	4.92	30	.045	7.46
10	Climb	3.358	4.88	1.894	5.2	0.352	2.35	31	.122	6.64
11	Forward Level Flight	0.1 V.H.	0.1 V.H.	0.2 V.H.	1.263	5.0	4.06	32	.110	11.15
	0.4 V.H.	0.4 V.H.	0.4 V.H.	2.526	4.3	5.90	33	.110	10.20	
	0.5 V.H.	0.5 V.H.	0.5 V.H.	2.526	5.4	6.49	34	.122	6.62	
	0.6 V.H.	0.6 V.H.	0.6 V.H.	3.473	5.3	0.840	4.81	35	.122	6.83
	0.7 V.H.	0.7 V.H.	0.7 V.H.	3.789	5.2	0.473	3.8	36	.100	7.54
	0.8 V.H.	0.8 V.H.	0.8 V.H.	12.628	5.3	0.316	4.5	37	.028	8.45
	0.9 V.H.	0.9 V.H.	0.9 V.H.	18.943	5.4	0.631	5.0	38	.014	6.11
	1.0 V.H.	1.0 V.H.	1.0 V.H.	9.471	5.3	0.631	4.6	39	.014	6.6
	Sideslip			0.800	8.94/5.17	0.868	3.5	40	.225	4.29
12	Autorotation									
13	Part Power Descent									
14	Power Dive 1.2 V.H.	2.000	5.29	1.068	5.29	0.267	4.87	41	.014	14.26
15	Take-Off and Climb	1.859	4.59	2.000	5.20	0.560	4.49	42	.003	10.21
16	Left Hover Turn	0.532	6.0	1.859	4.59	0.465	3.36	43	.2	9.85
17	Right Hover Turn	0.440	6.42	0.532	6.0	0.133	5.75	44	.1	9.41
18	30° A.O.B. Left Th.	132	0.440	0.440	7.63	0.110	5.69			
19	30° A.O.B. Right Th.	600	3.333	3.333	5.17	150	0.833	5.54		
20	30° A.O.B. Right Th.	600	3.333	3.333	5.17	150	0.833	6.05/3.05		
21	45° A.O.B. Left Th.	128	0.532	0.532	5.67	32	0.133	6.02		
22	45° A.O.B. Right Th.	128	0.532	0.532	5.71	32	0.133	5.90		
23	60° A.O.B. Left Th.	28	0.055	6.43	7	0.014	7.47			
24	60° A.O.B. Right Th.	28	0.055	6.12	7	0.014	7.54			
25	Auto Left Turn	40	0.167	4.86	15	0.042	4.99			
26	Auto Right Turn	40	0.167	4.60	15	0.042	4.89			
27	Hover Approach	400	0.446	5.40	100	.111	5.37			

TABLE II. GIMBAL SPRING SIZING

<u>Material</u>	<u>Thickness, In.</u>			<u>Weight, Lb</u>
	<u>Root</u>	<u>35%</u>	<u>Tip</u>	
Fiberglass	1.53	0.95	0.28	96.2
Graphite	1.01	0.63	0.19	65.1

TABLE III
VALUES USED IN MERIT FUNCTIONS

PARAMETER	ROTOR HUB GOAL	EGR	I-BEAM TORQUE TUBE	ARTIC- COMPOS. ELAST.	UH-60A PROD.
DRAG	2.8 ft ²	1.6	3.2	3.4	2.0
WEIGHT	400 lb	433	389	456	570
PARTS COUNT	50	31	38	55	75
VULNERABLE AREA	0 ft ²	1.63	1.66	2.06	-
HUB MOMENT STIFFNESS	120,000 TC 180,000 ft-lb rad	139,000	175,000	150,000	175,000

TABLE IV
ITR MERIT FACTORS

NOTE: All Merit Factors Were Calculated Without Accounting For Blade Fold

	GOAL	ALLOW. RANGE	EGR	I-BEAM TORQUE TUBE	STIFFENED I-BEAM TORQUE TUBE	ARTIC. COMPOS. ELAST.	UH-60A PROD. HUB
VULNERABILITY - K_v	-	0 to 1	0.90	0.90	0.90	0.87	-
AEROMECH. STAB. - K_a	-	0 to 1	0.75	0.85	0.85	1.00	1.00
DRAG - K_d	2.8 ft ²	-	42.9	-14.3	-14.3	-21.4	28.5
WEIGHT - K_w	400 lb	-	-8.2	2.8	-2.2	-14.0	-42.6
PARTS COUNT - K_p	50	-	38.0	24.0	24.0	-10.9	-50.0
HUB MOMENT STIFF. - K_e	150,000 ft-lb	\leq 5 rad	5.0	5.0	-0.9	5.0	5.0
HUB MOMENT - K_m	10,000 ft-lb	-	12.5	0	80.8	80.8	42.9
TIP PATH PLANE TILT ANGLE - K_b	5 deg	-	5.9	-17.2	2.4	35.7	10.8
RELIABILITY - K_r	3000 hr MTBR	0 to 10	2.0	3.0	3.0	3.0	0
MANUFACTURING COST - K_c	-	1 to 10	9.0	8.0	8.0	8.0	7.0
FATIGUE LIFE - K_f	10,000 hr	0 to 10	5.0	2.0	10.0	10.0	0
AUX. LEAD-LAG DAMPER - K_z	-	0 to 2	1.5	1.5	1.5	2.0	2.0
TORS. STIFFNESS - K_s	\leq 150% CUR. LDS	2 to 0	0	0	0	0	0
TOTAL			113.6	14.8	112.3	99.1	3.6
TOTALS X K_a			85.2	12.6	95.5	99.1	3.6
TOTALS X K_a X K_v			76.7	11.3	85.9	86.2	-

TABLE V
MODIFIED ITR MERIT FACTORS

NOTE: All Merit Factors Were Calculated Without Accounting For Blade Fold

FACTOR	ROTOR HUB GOAL	ALLOW. RANGE	MERIT MULTIPLIER	EGR	I-BEAM TORQUE TUBE	STIFFENED I-BEAM TORQUE TUBE	ARTIC. COMPOS. ELAST.	UH-60A PROD. HUB
VULNERABILITY - K_v	-	0 to 1	1.0	0.90	0.90	0.90	0.87	-
AEROMECH. STAB. - K_a	-	0 to 1	1.0	0.75	0.85	0.85	1.00	1.00
DRAG - K_d	2.8 ft ²	-	0.20	-8.6	-2.9	-2.9	-4.3	5.7
WEIGHT - K_w	400 lb	-	0.6	-4.9	1.7	-1.3	-8.4	-25.6
PARTS COUNT - K_p	50	-	0.15	5.7	3.6	3.6	-1.5	-7.5
HUB MOMENT STIFFNESS - K_e	150,000 ft-lb	5	2.0	10.0	10.0	-1.8	10.0	10.0
HUB MOMENT - K_m	10,000 ft-lb	-	0.1	1.3	0	8.1	8.1	4.3
TIP PATH PLANE TILT ANGLE - K_b	5 deg	-	0.1	0.6	-1.7	0.2	3.6	1.1
RELIABILITY - K_r	3,000 hr	0 to 10	1.0	2.0	3.0	3.0	3.0	0
MANUFACTURING COST - K_c	-	1 to 10	1.0	9.0	8.0	8.0	8.0	7.0
FATIGUE LIFE - K_f	10,000 hr	0 to 10	1.0	5.0	2.0	10.0	10.0	0
AUX. LEAD-LAG DAMPER - K_z	-	0 to 2	1.0	1.5	1.5	1.5	2.0	2.0
TORS. STIFF. - K_s	150% CUR LDS	2 to 0	1.0	0	0	0	0	0
TOTALS				38.8	25.2	28.4	17.0	-3.0
TOTALS X K_a				29.1	21.4	24.1	17.0	-3.0
TOTALS X K_a X K_v				26.2	19.3	21.7	14.8	-

TABLE VI

HUB DRAG BREAKDOWN

HUB CONFIGURATION	DRAG FACTORS	PROJECTED AREA FT ²	EQUIV. DRAG COEFF.	HUB DRAG FT ²	TORQUE TUBE EQ. DRAG INCREMENT FT ²	TOTAL HUB DRAG FT ²
EGR		4.75	.331	1.6	0	1.6
I-BEAM/TORQUE TUBE		1.58	.513	0.8	2.4*	3.2
ARTIC. COMPOSITE ELASTOMERIC		3.69	.921	3.4	0	3.4
UH-60A PROD.		2.73	.744	2.0	0	2.0

*THIS IS THE EQUIVALENT DRAG AREA INCREMENT RELATIVE TO A UH-60A ROTOR BLADE



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TABLE VII
EGR AND I-BEAM/TORQUE TUBE ROTOR DRAG BREAKDOWN

<u>ITEM</u>	<u>EGR</u>	<u>I-BEAM</u>
Hub fairing swept area	5.22 ft ²	1.92 ft ²
Hub fairing pressure drag coefficient	0.272	0.412
Hub fairing skin friction drag coefficient	0.0205	0.0105
Hub fairing f (1)	1.53 ft ²	0.811 ft ²
Blade shank swept area	0.159 ft ²	--
Blade shank C_D	0.0064	--
Blade shank f (2)	0.033 ft ²	2.42 ft ² *
Total f (1) + (2)	1.563 ft ²	3.231 ft ²

*Computed using the Generalized Rotor Performance (GRP) program

NOTE: all C_D values are non-dimensionalized with respect to hub swept area.

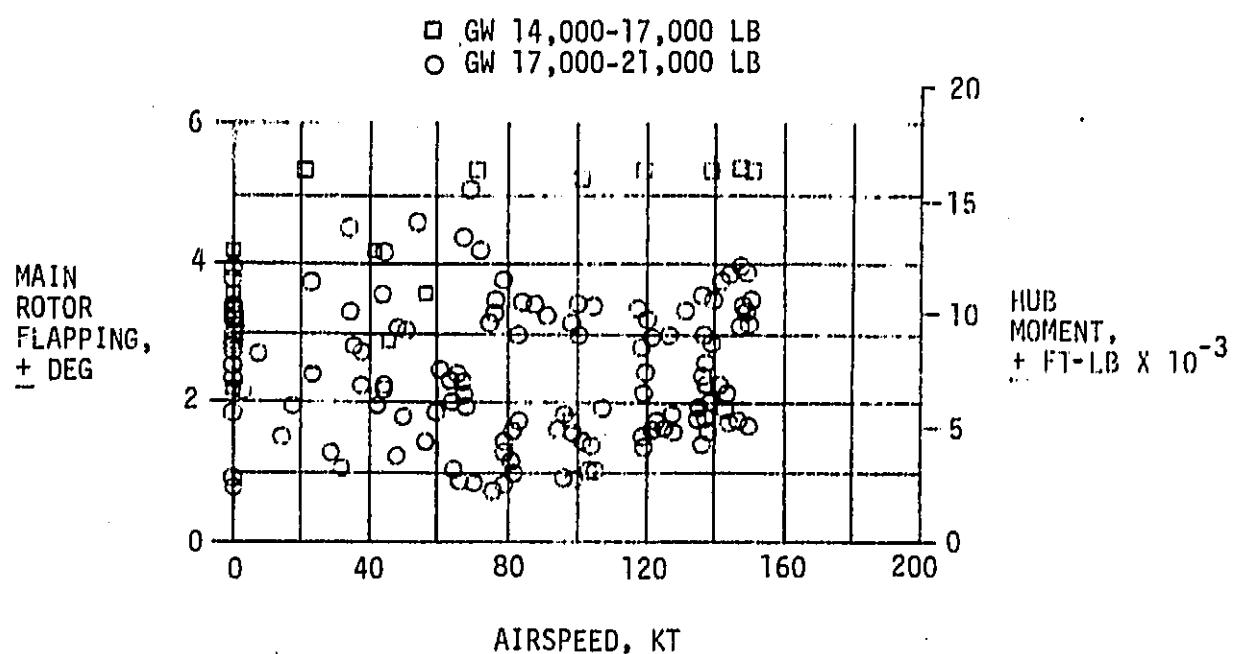


FIGURE 1. UH-60A MAIN ROTOR FLAPPING/HUB MOMENT.

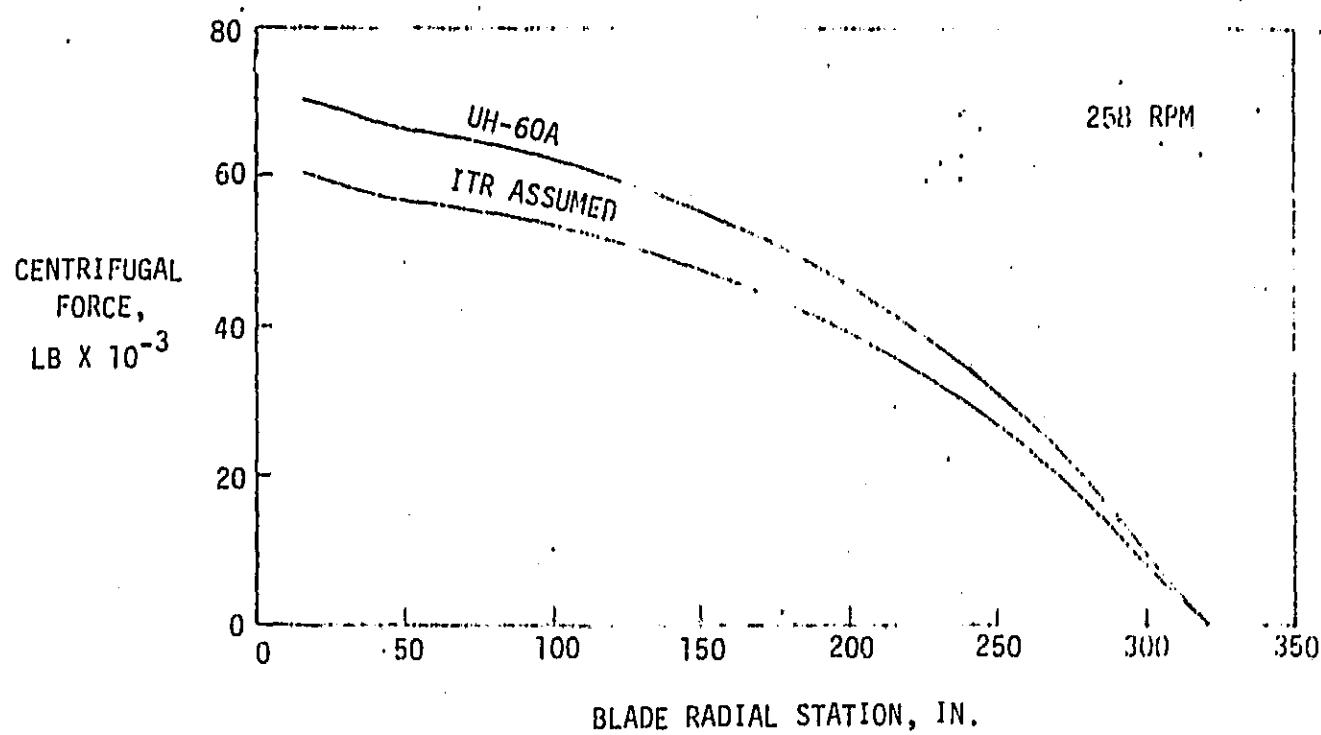


FIGURE 2. CENTRIFUGAL FORCE RADIAL DISTRIBUTION.

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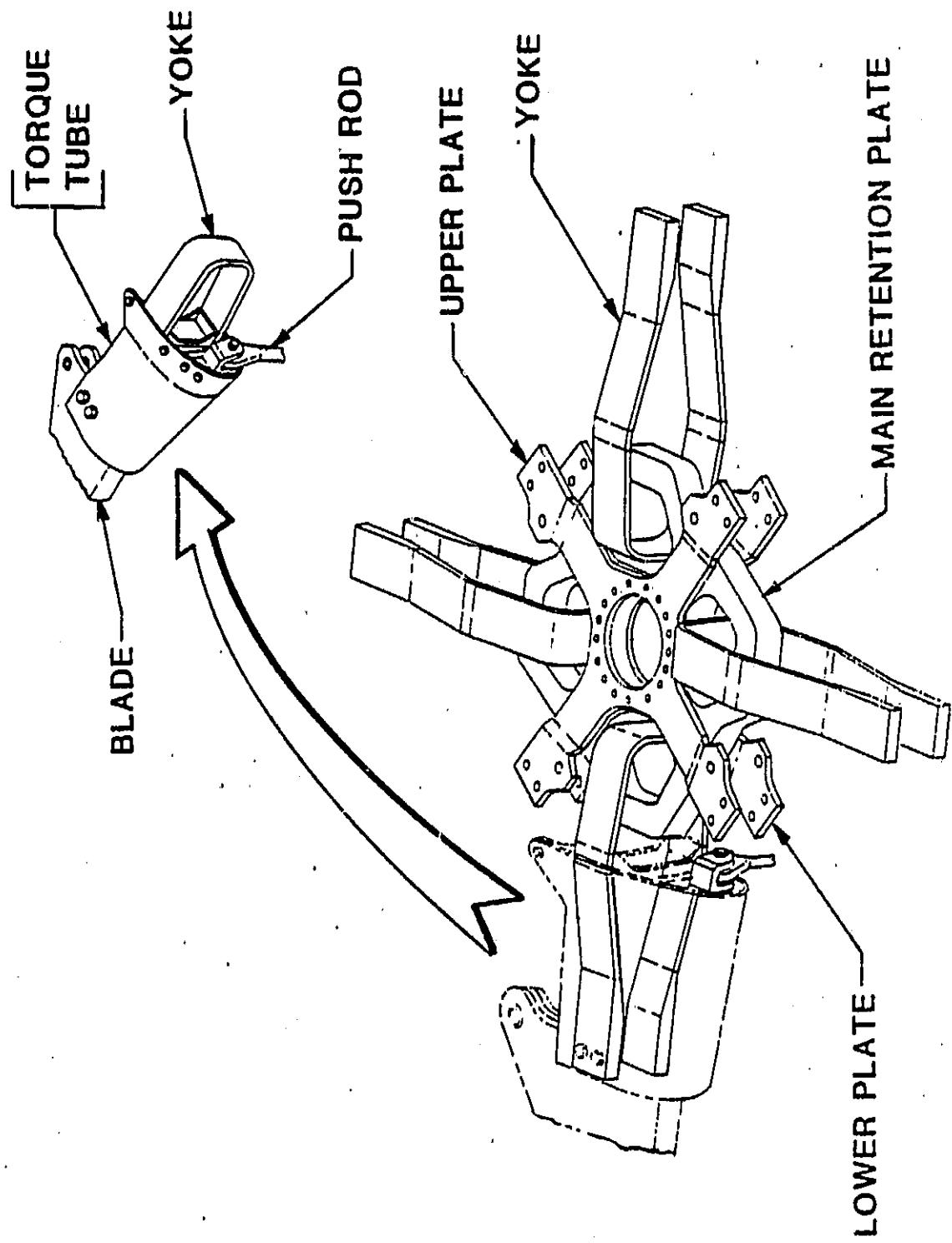


FIGURE 3. ARTICULATED COMPOSITE-ELASTOMERIC HEAD, ISOMETRIC.

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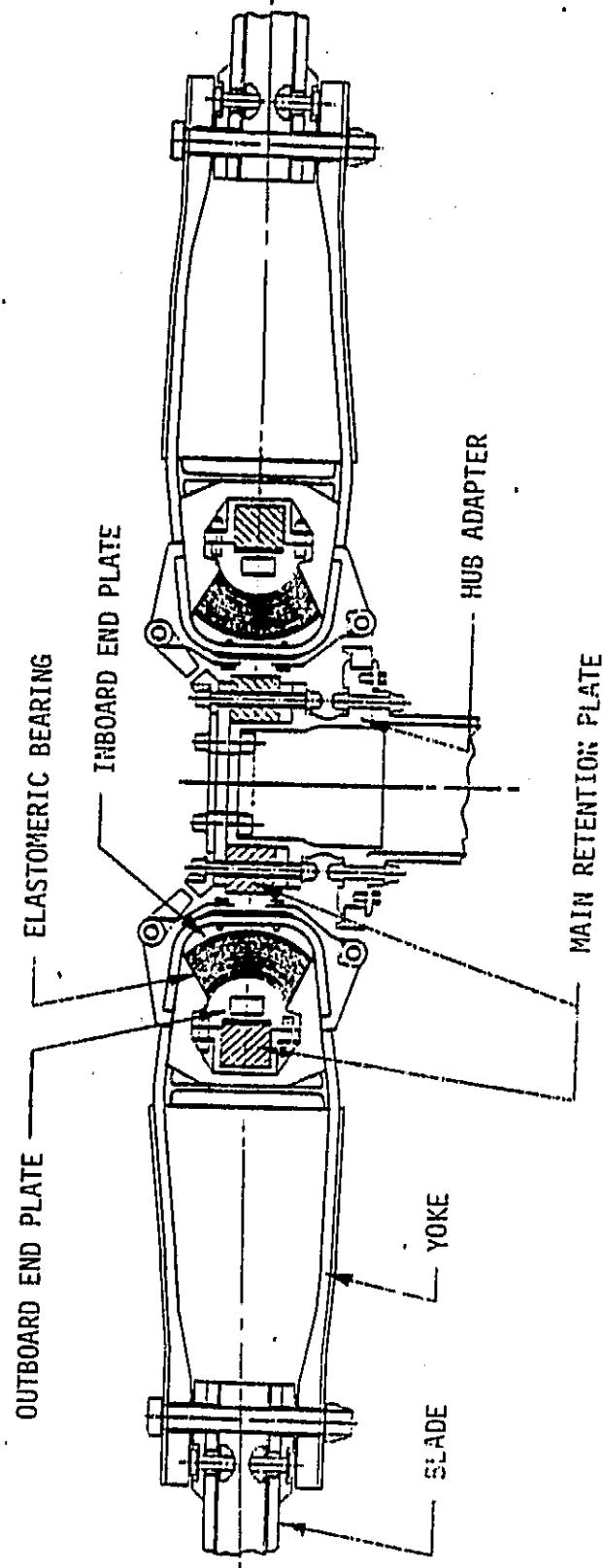


FIGURE 4. ARTICULATED COMPOSITE-ELASTOMERIC HEAD, CROSS-SECTION.

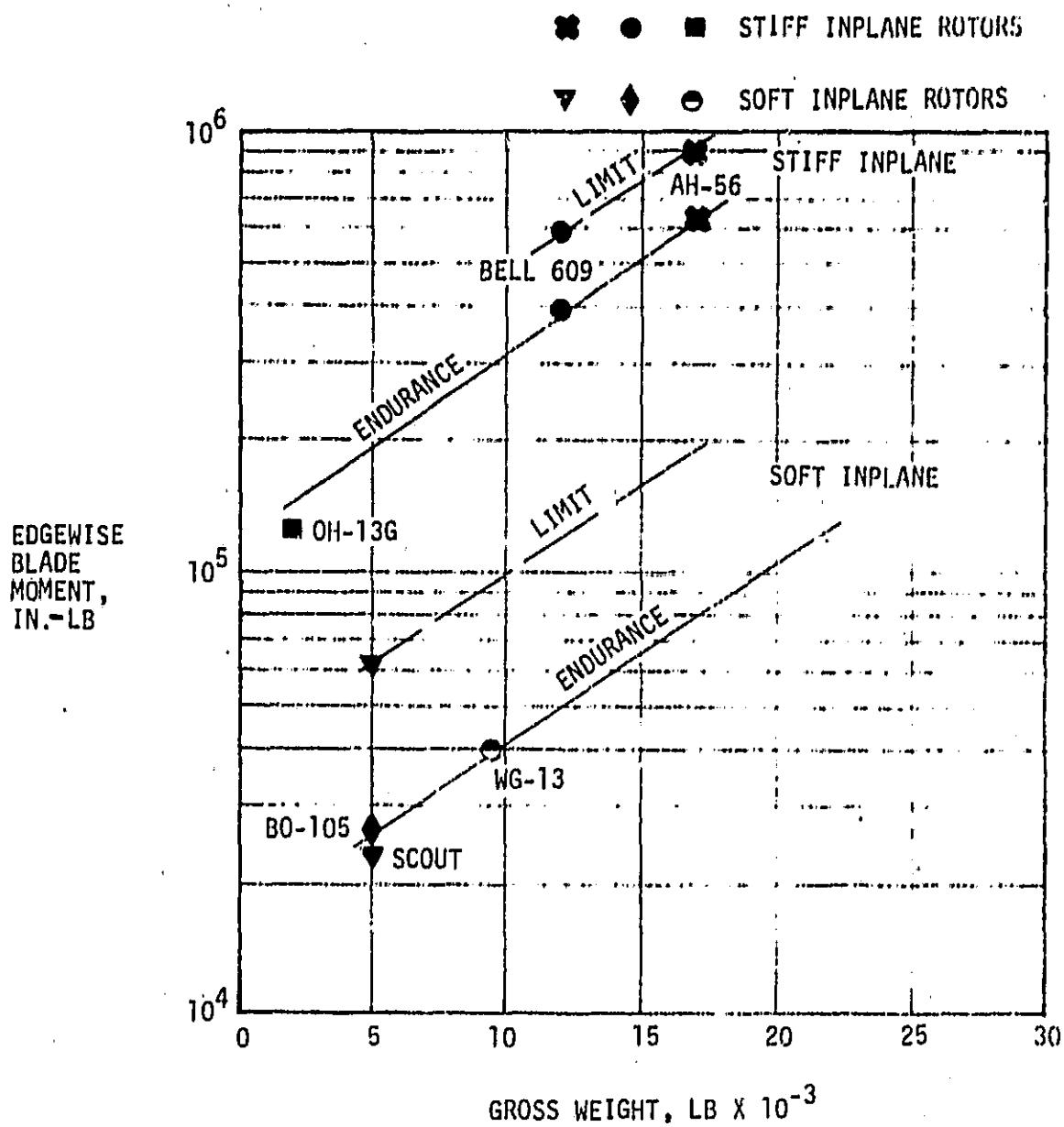


FIGURE 5. DESIGN EDGEWISE ROOT BENDING MOMENTS FOR VARIOUS HINGELESS ROTORS.

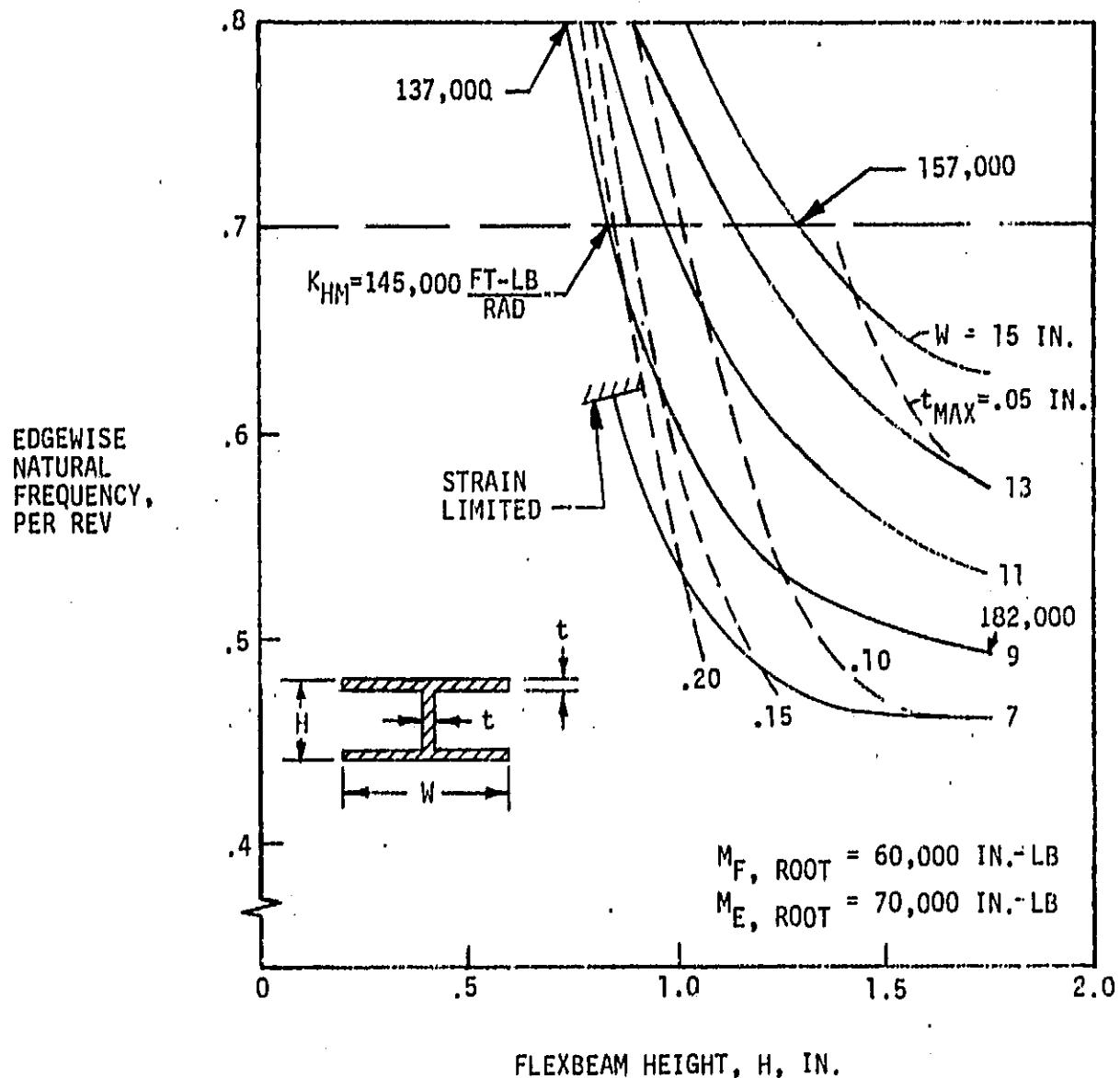


FIGURE 6. EFFECTS OF GRAPHITE I-BEAM WIDTH AND HEIGHT ON EDGEWISE FREQUENCY.

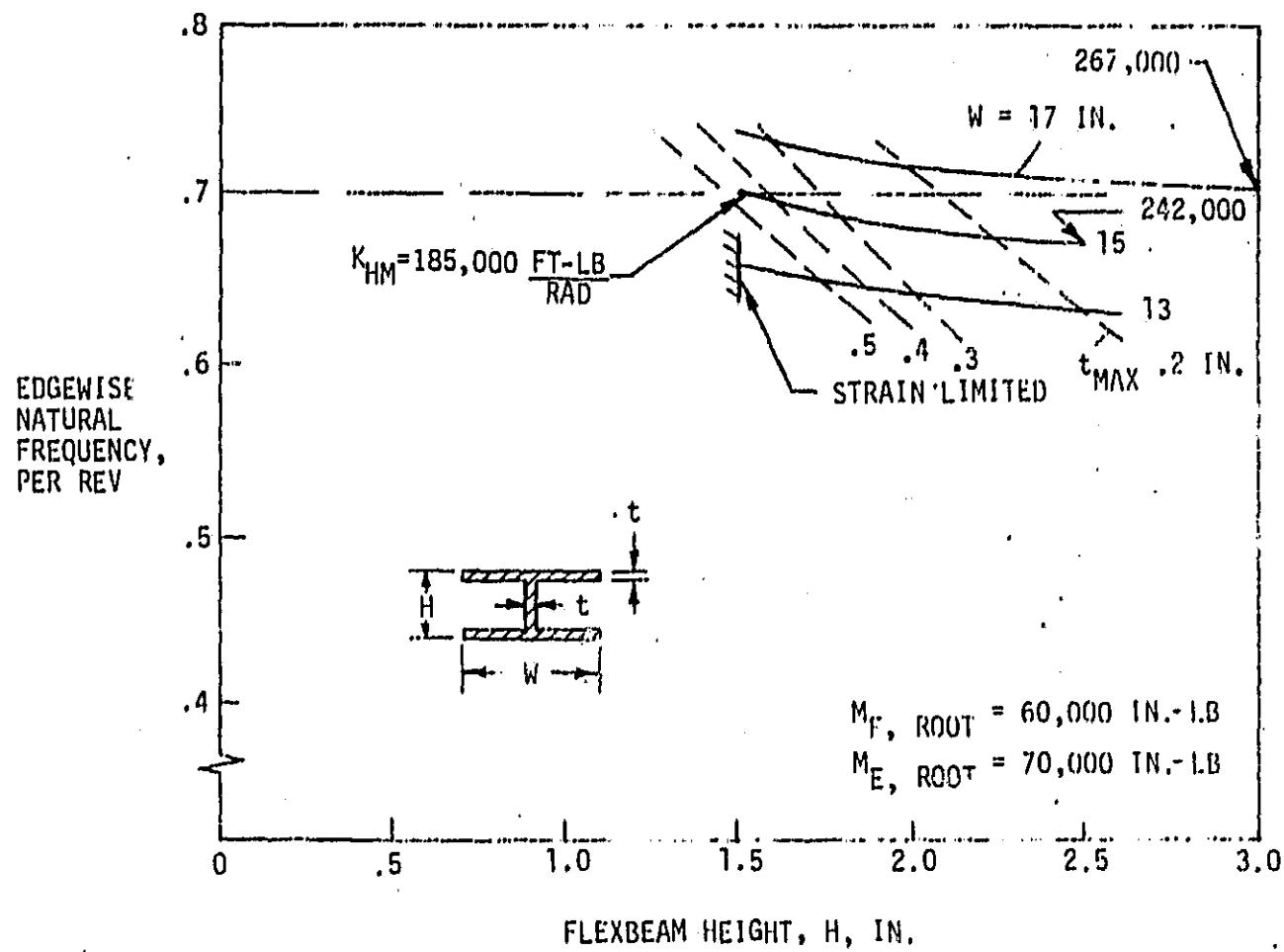


FIGURE 7. EFFECTS OF FIBERGLASS I-BEAM WIDTH AND HEIGHT ON EDGEWISE FREQUENCY.

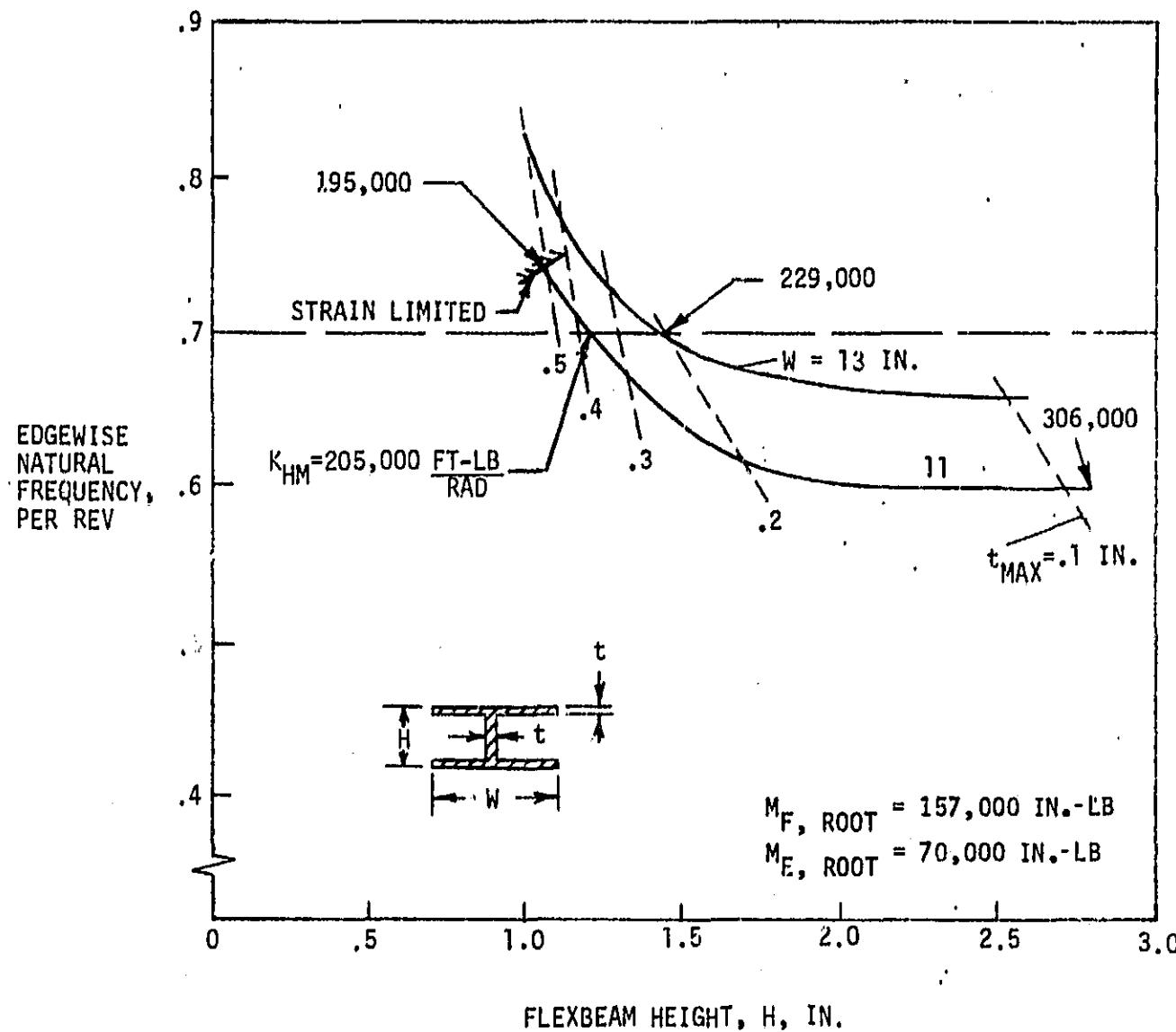


FIGURE 8. EFFECTS OF GRAPHITE I-BEAM WIDTH AND HEIGHT ON EDGEWISE FREQUENCY FOR INCREASED FLATWISE DESIGN MOMENT.

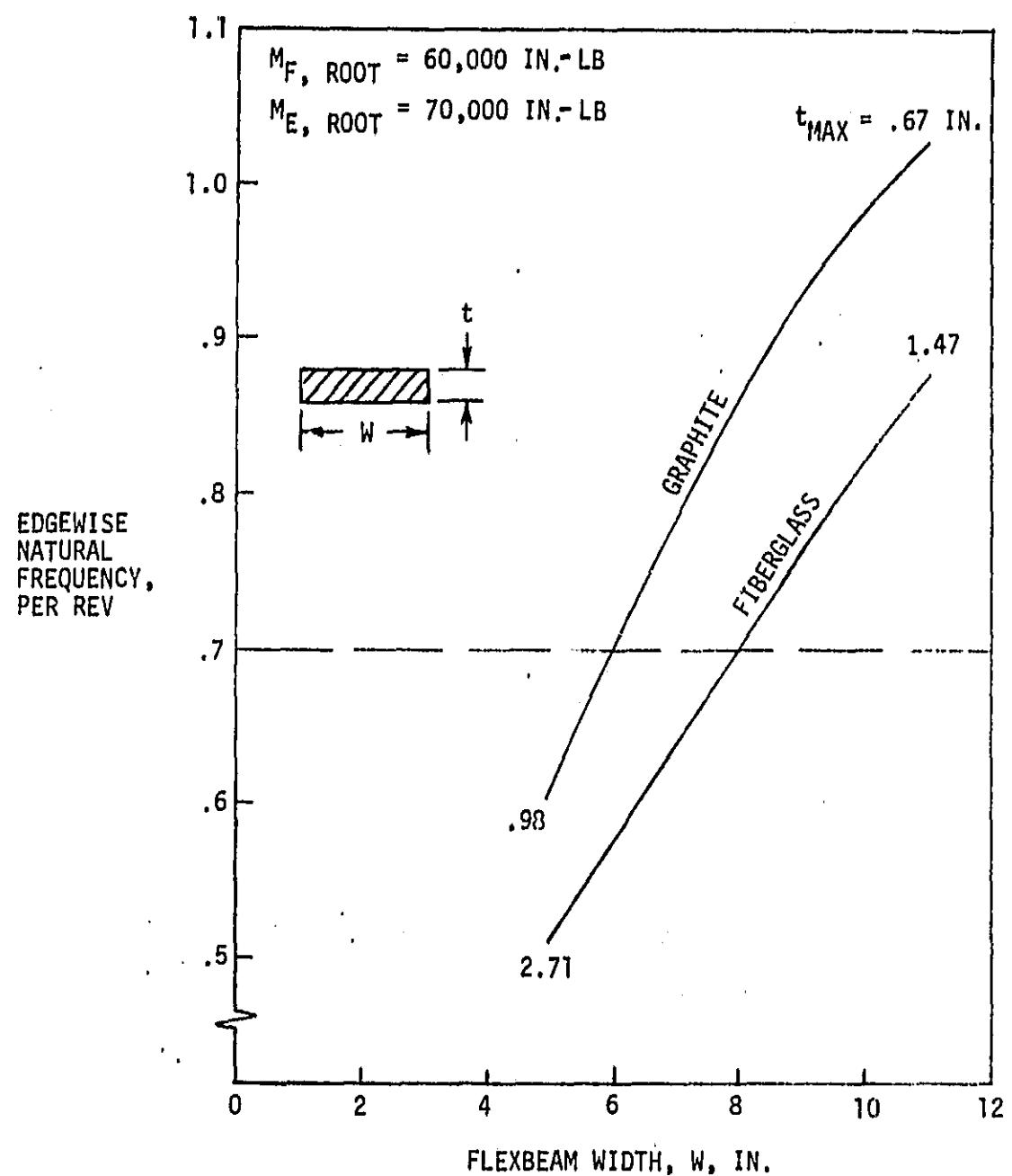


FIGURE 9. EFFECTS OF MATERIAL AND RECTANGULAR FLEXBEAM WIDTH ON EDGEWISE FREQUENCY.

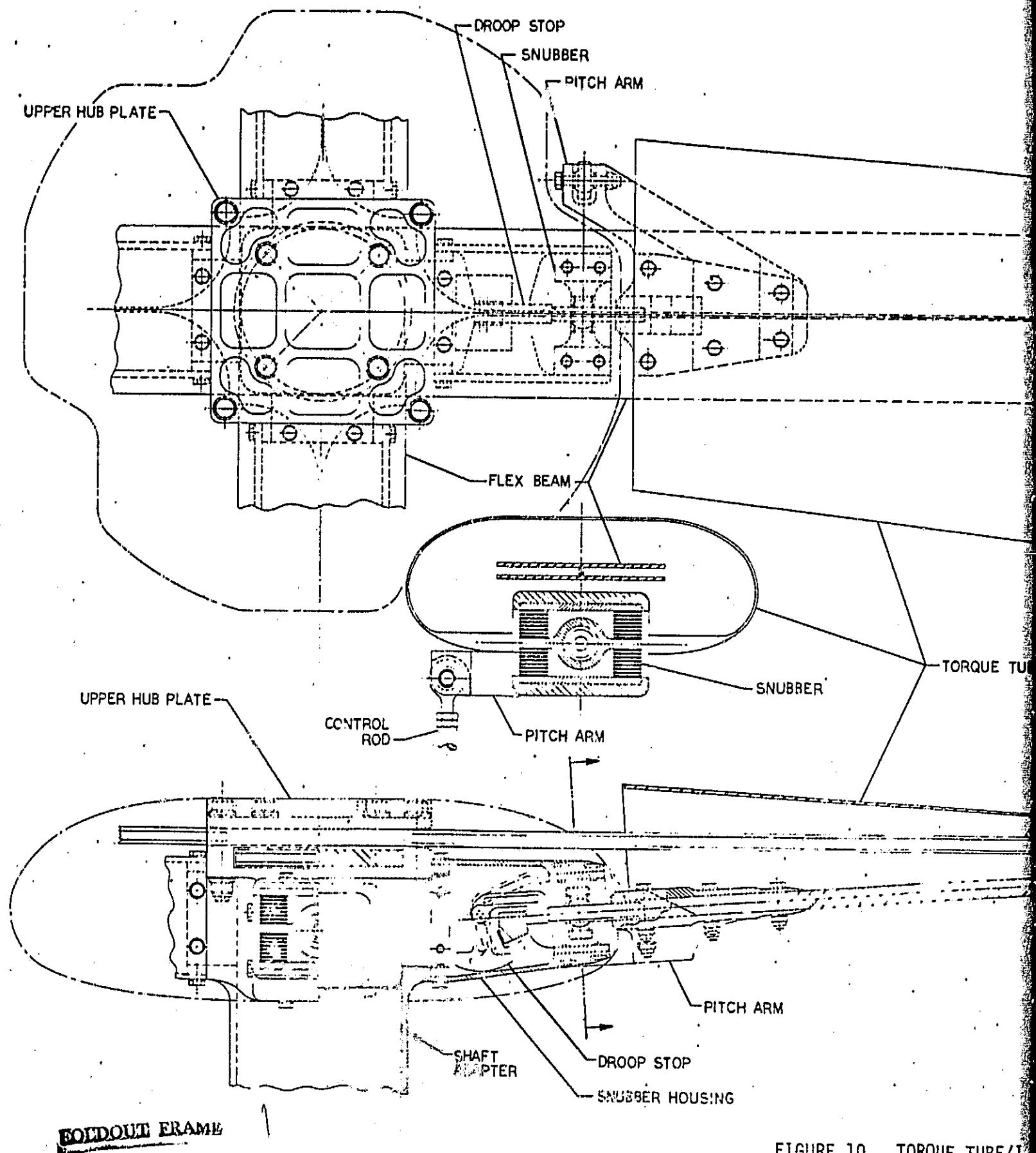
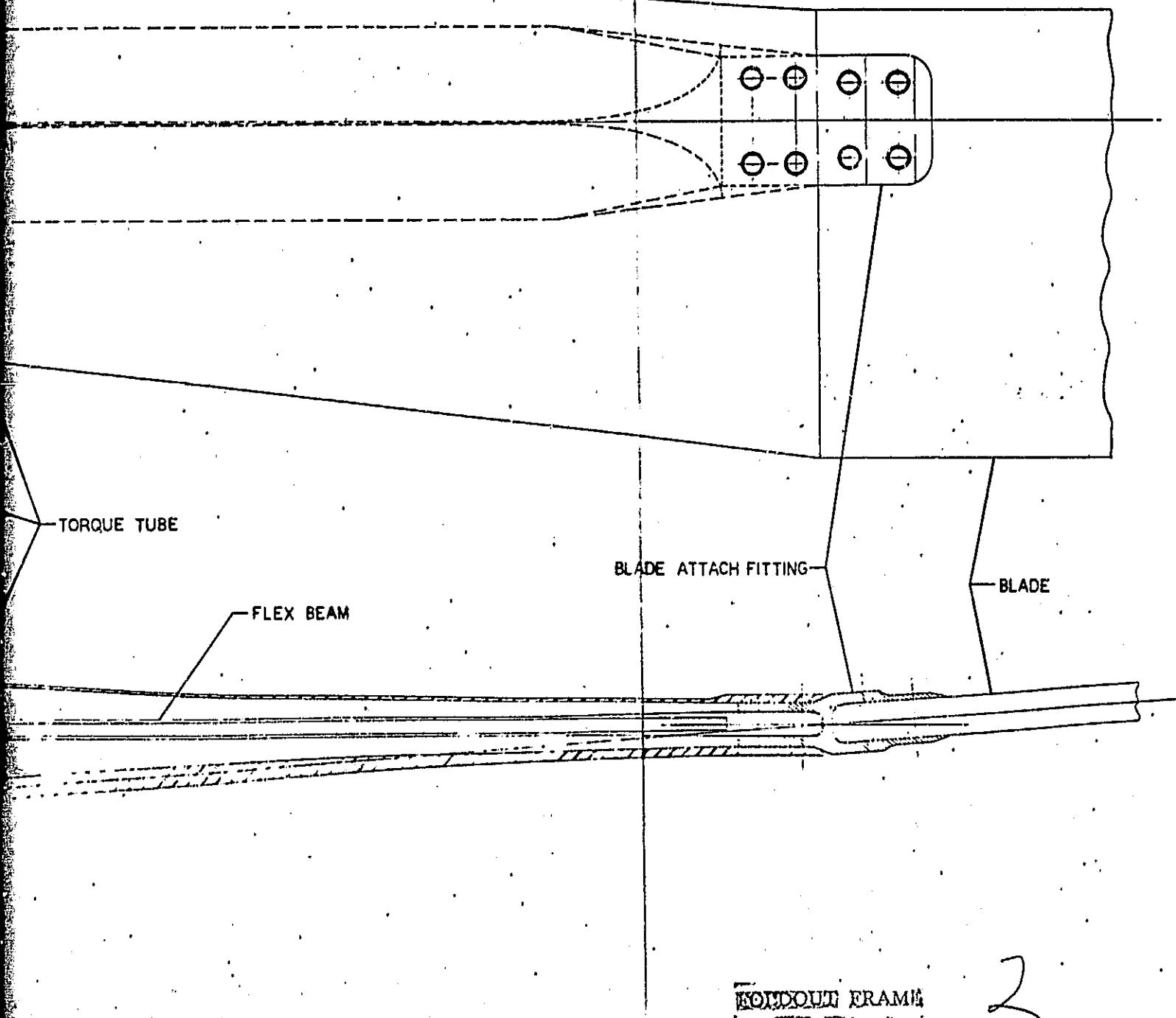


FIGURE 10. TORQUE TUBE/I

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TORQUE TUBE/I-BEAM ROTOR.

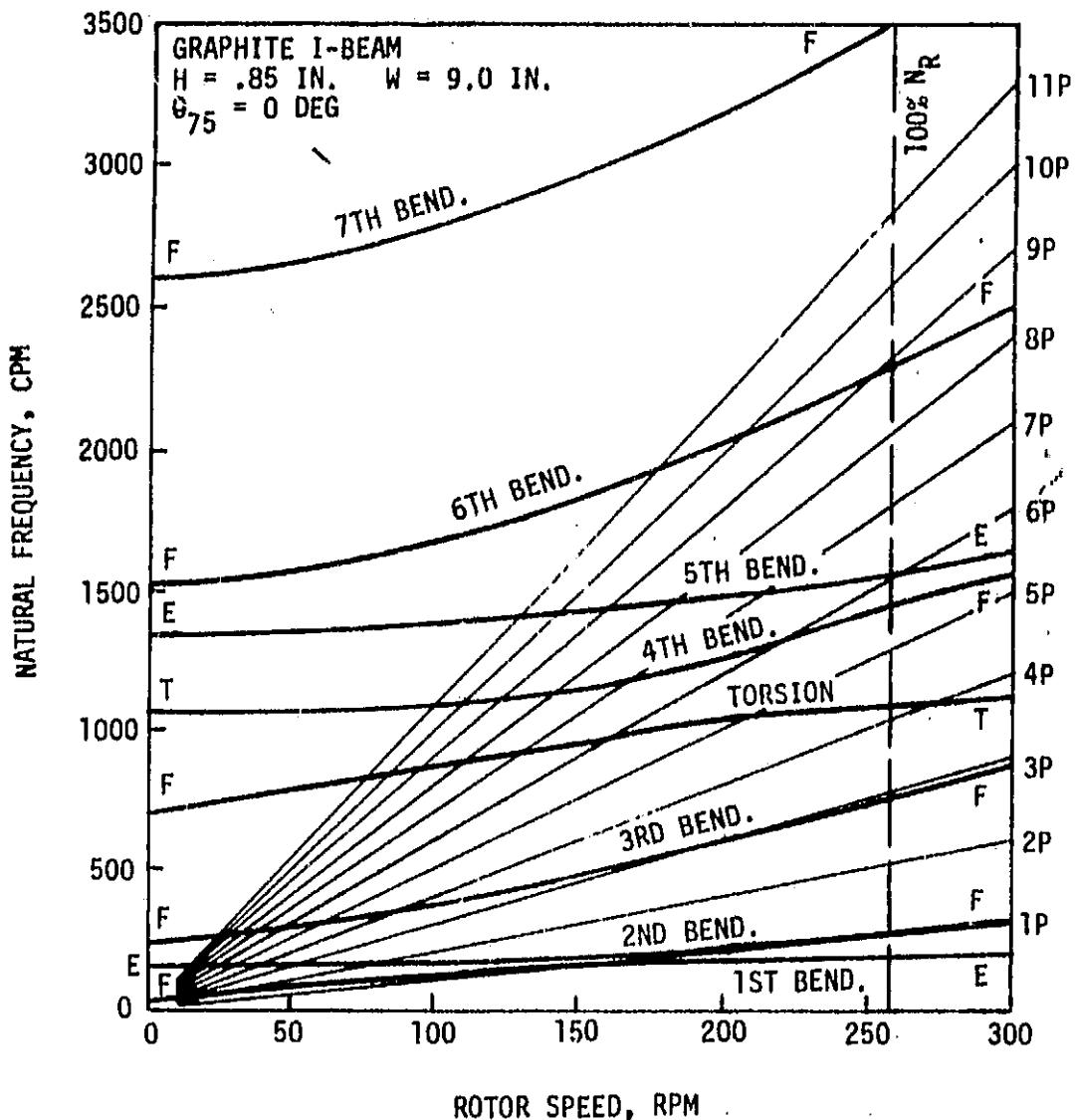


FIGURE 11. TORQUE TUBE/I-BEAM BEARINGLESS ROTOR FREQUENCIES.

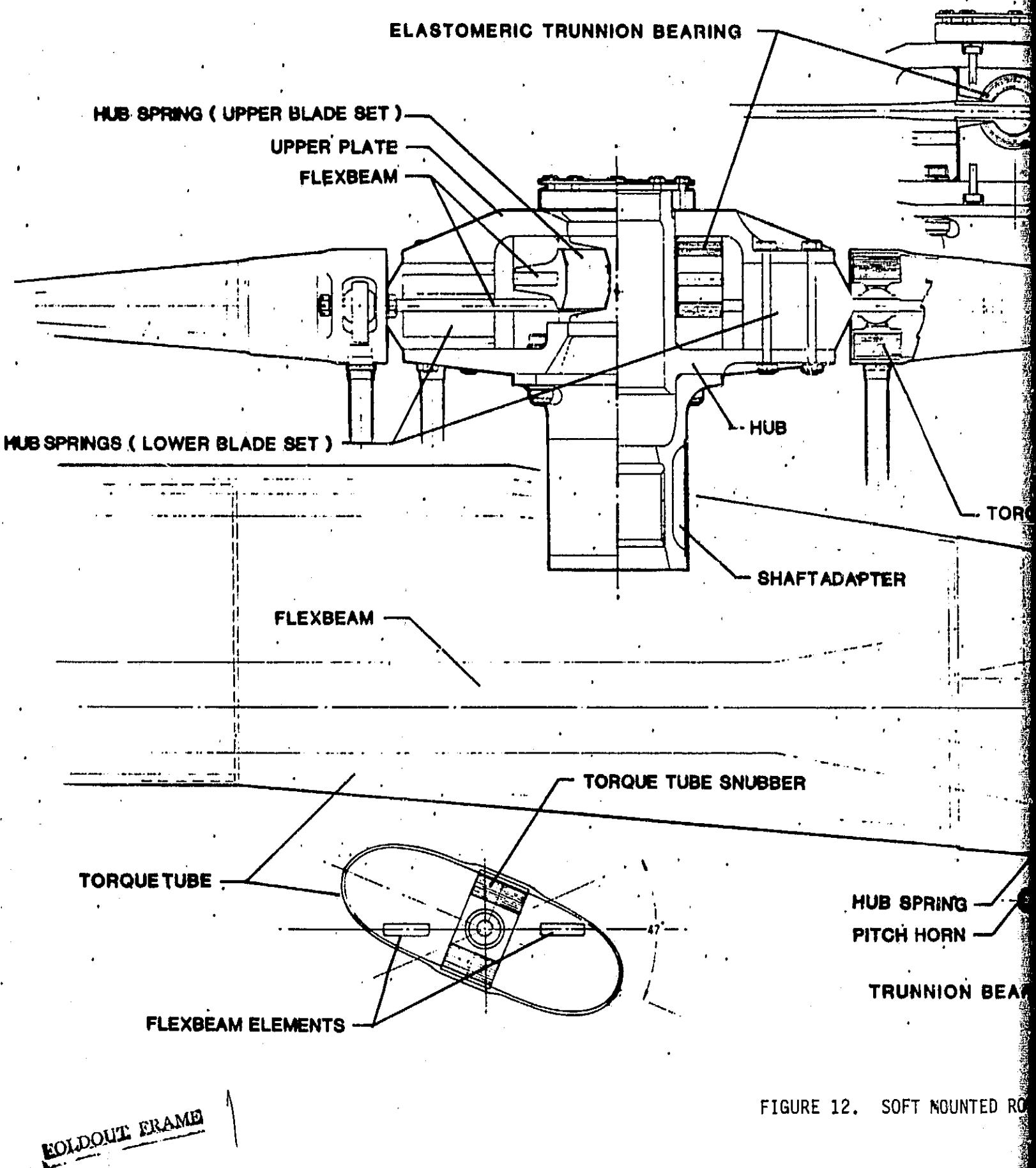
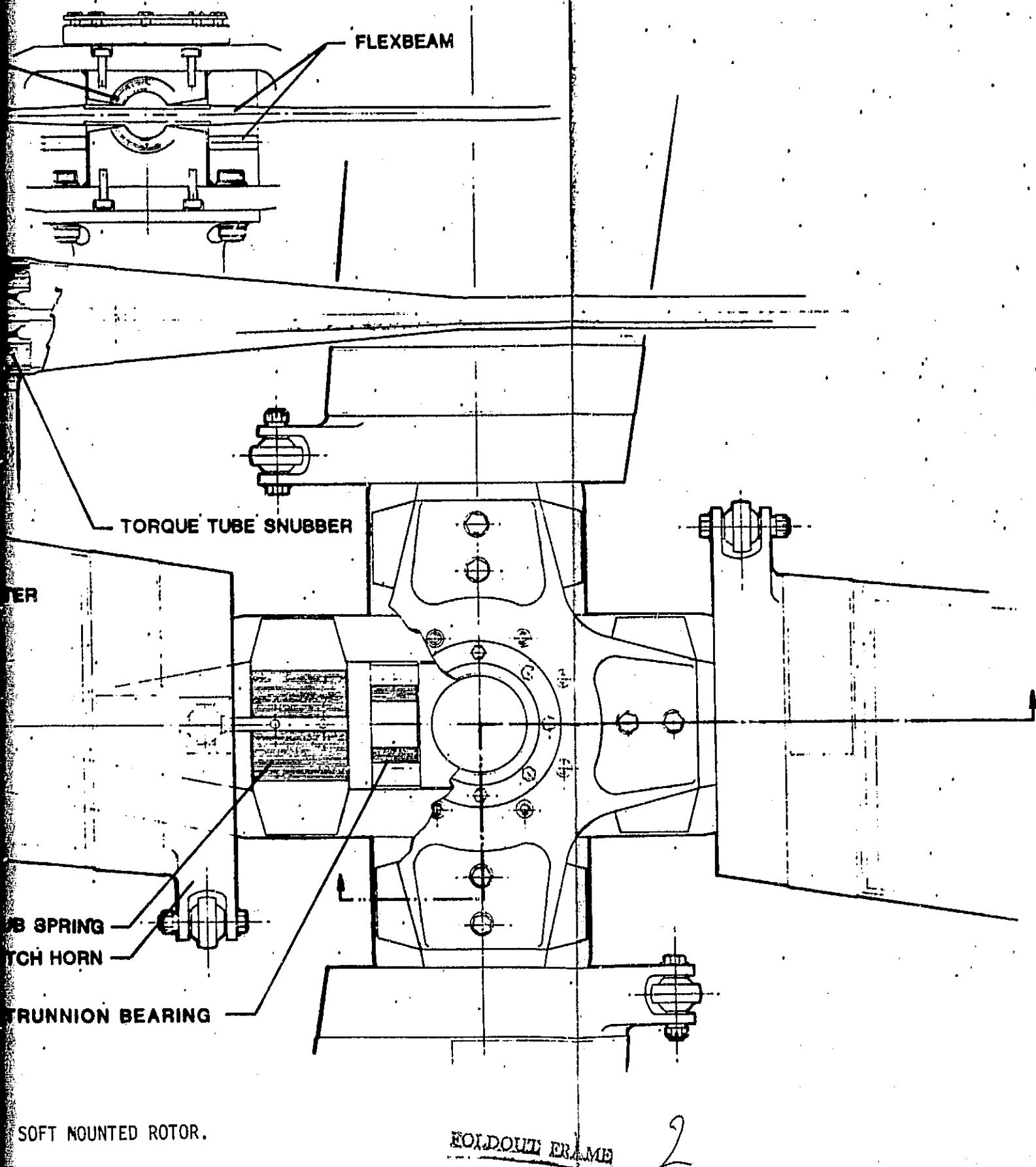


FIGURE 12. SOFT MOUNTED RO



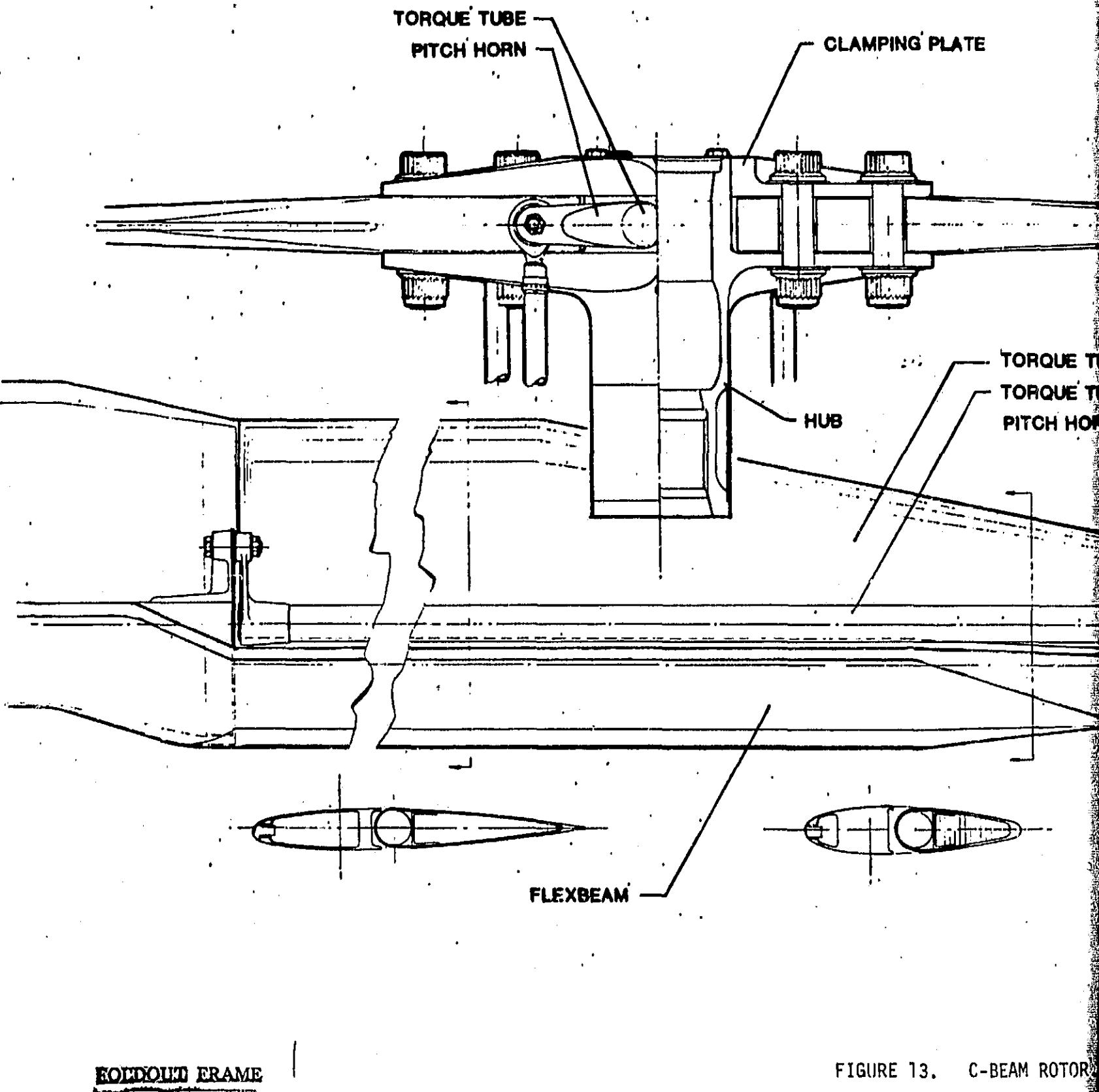
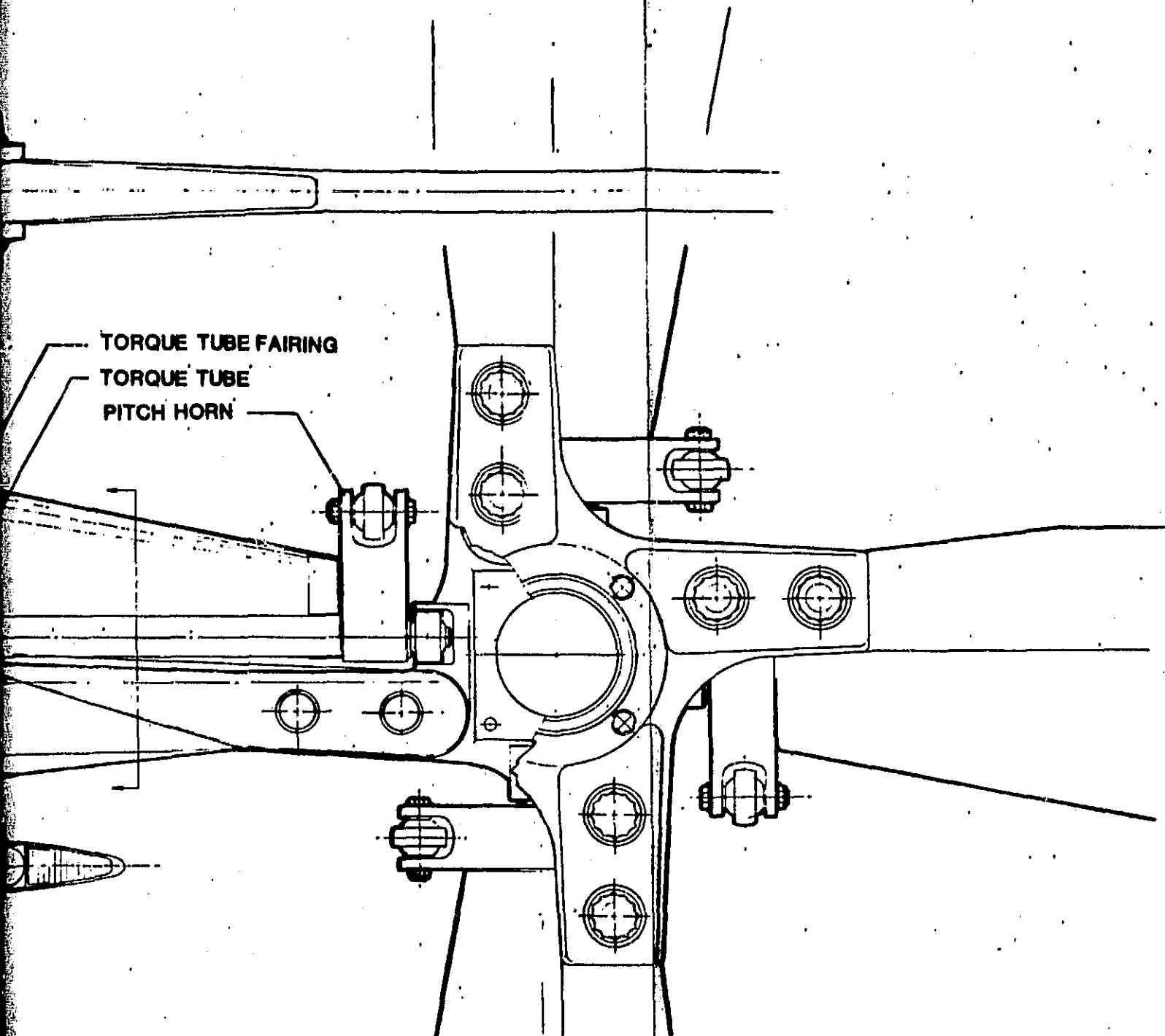


FIGURE 13. C-BEAM ROTOR

PLATE



3. C-BEAM ROTOR.

ENDOUE FRAME

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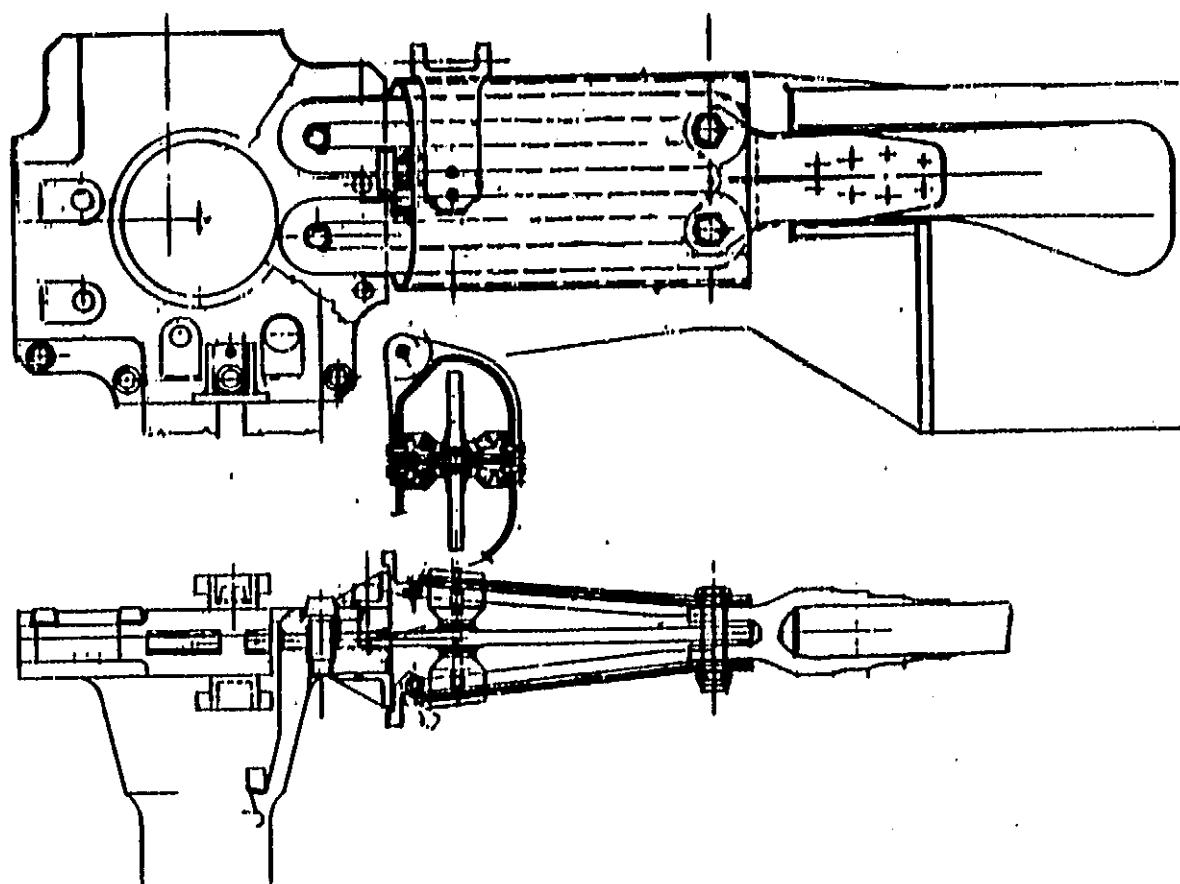
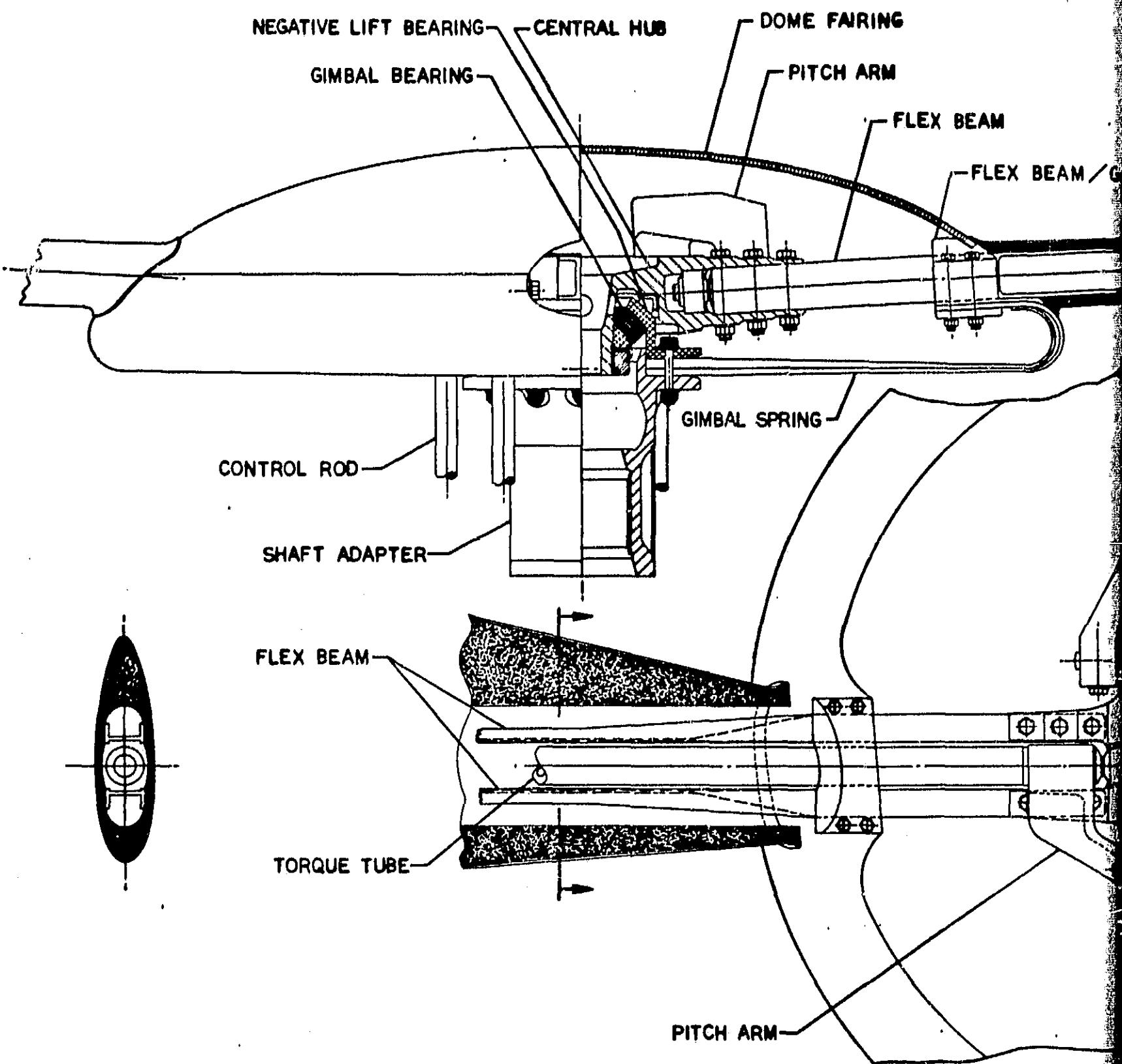


FIGURE 14. ANVIL-STRAP ROTOR.

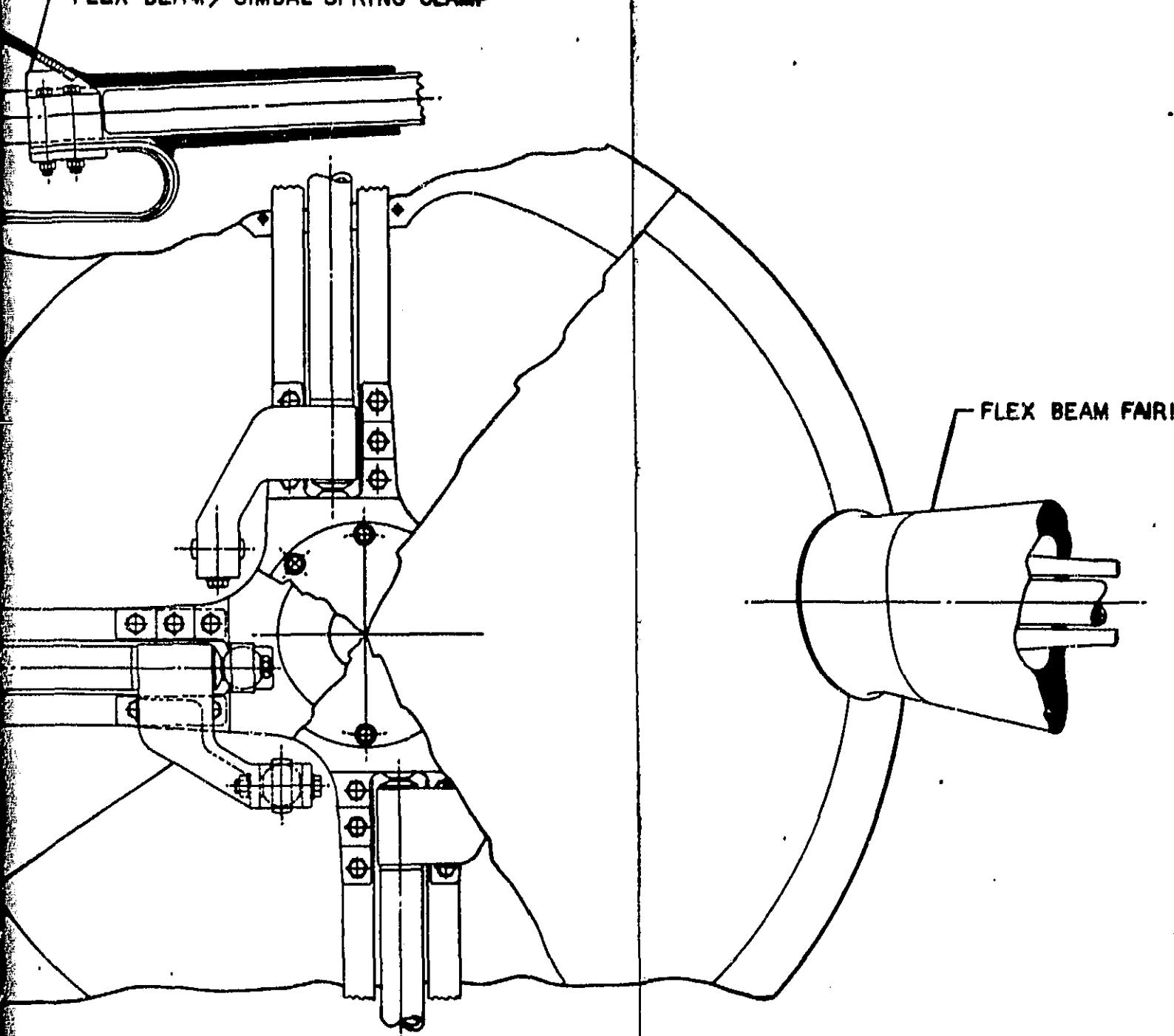


FOLDOUT FRAME

FIGURE 15. ELASTIC GIMBAL ROTOR

FLEX BEAM

FLEX BEAM/GIMBAL SPRING CLAMP



GIMBAL ROTOR

FOLDDOWN FRAME

2

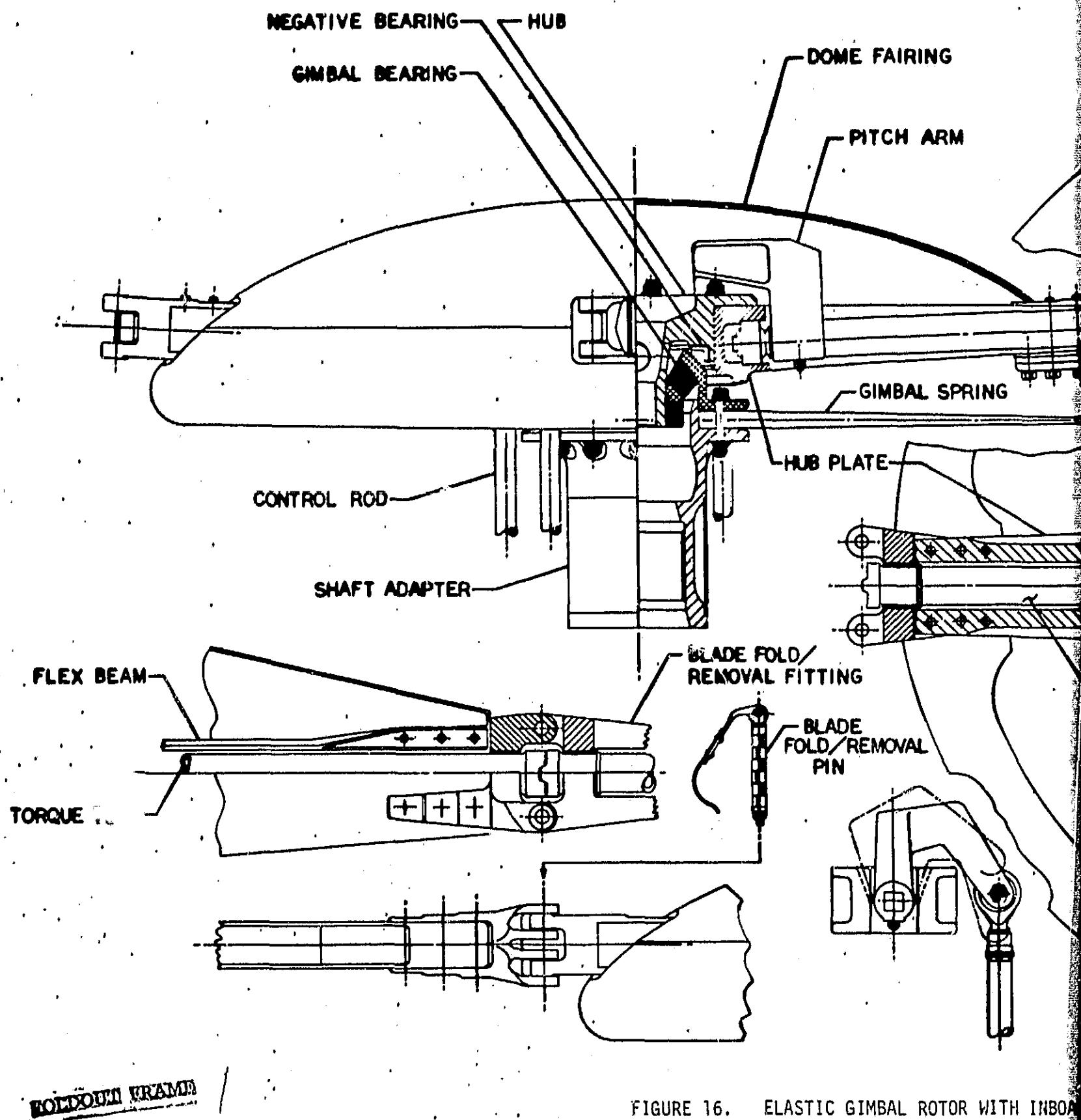
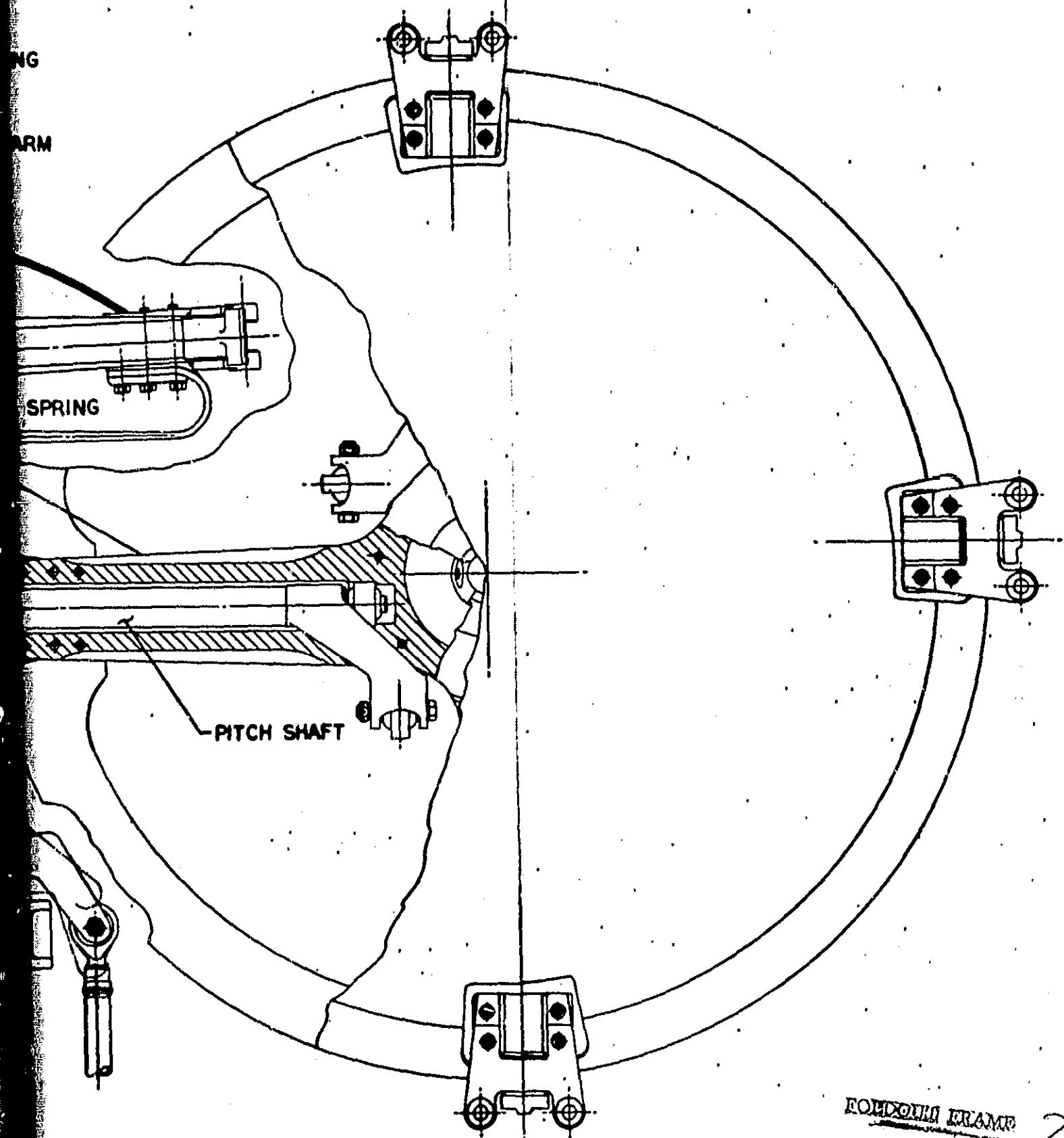


FIGURE 16. ELASTIC GIMBAL ROTOR WITH INBOARD FOLD

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OR WITH INBOARD BLADE FOLD

FOLDING FRAME

2

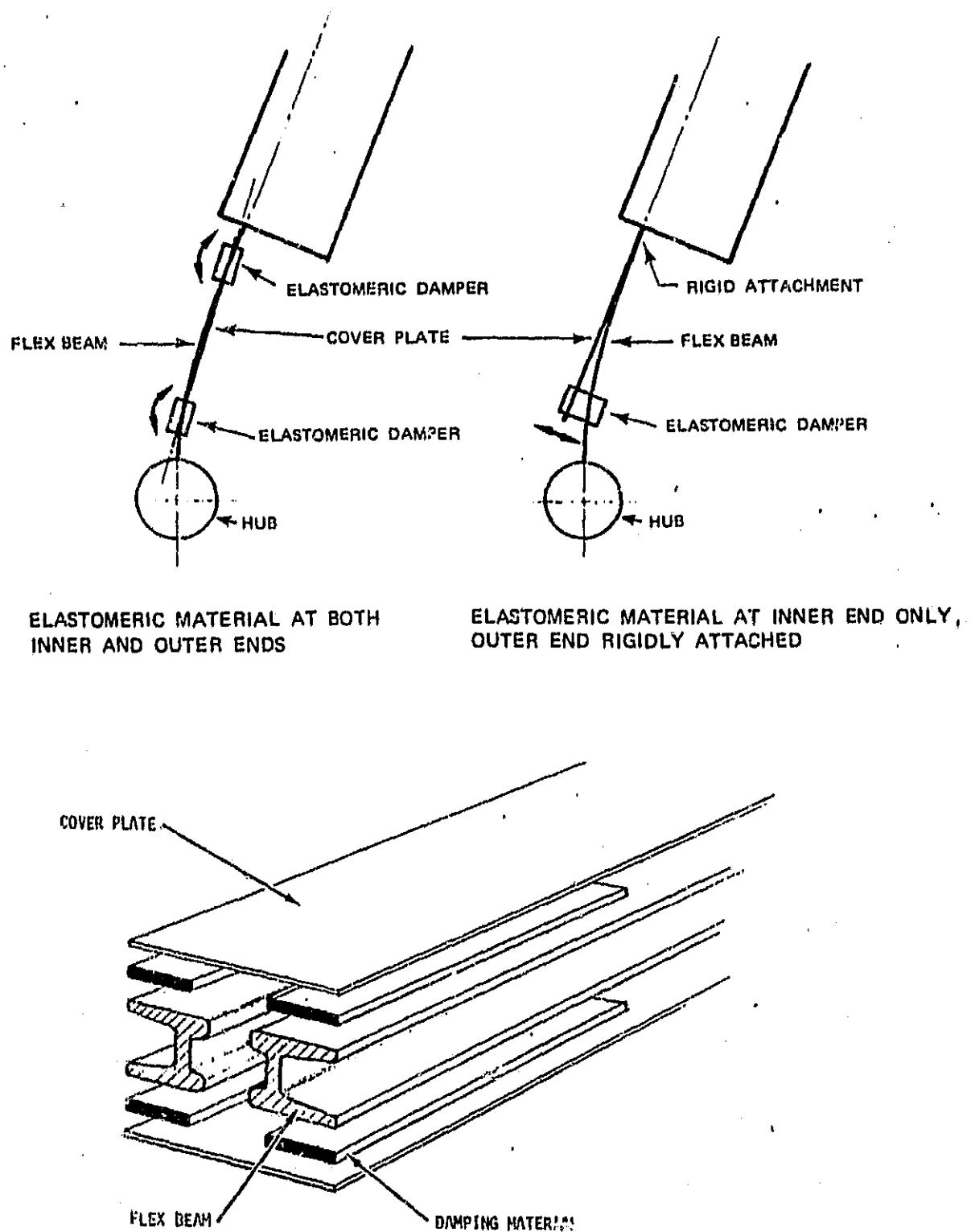


FIGURE 17. AUXILIARY LAG DAMPING FOR EGR.

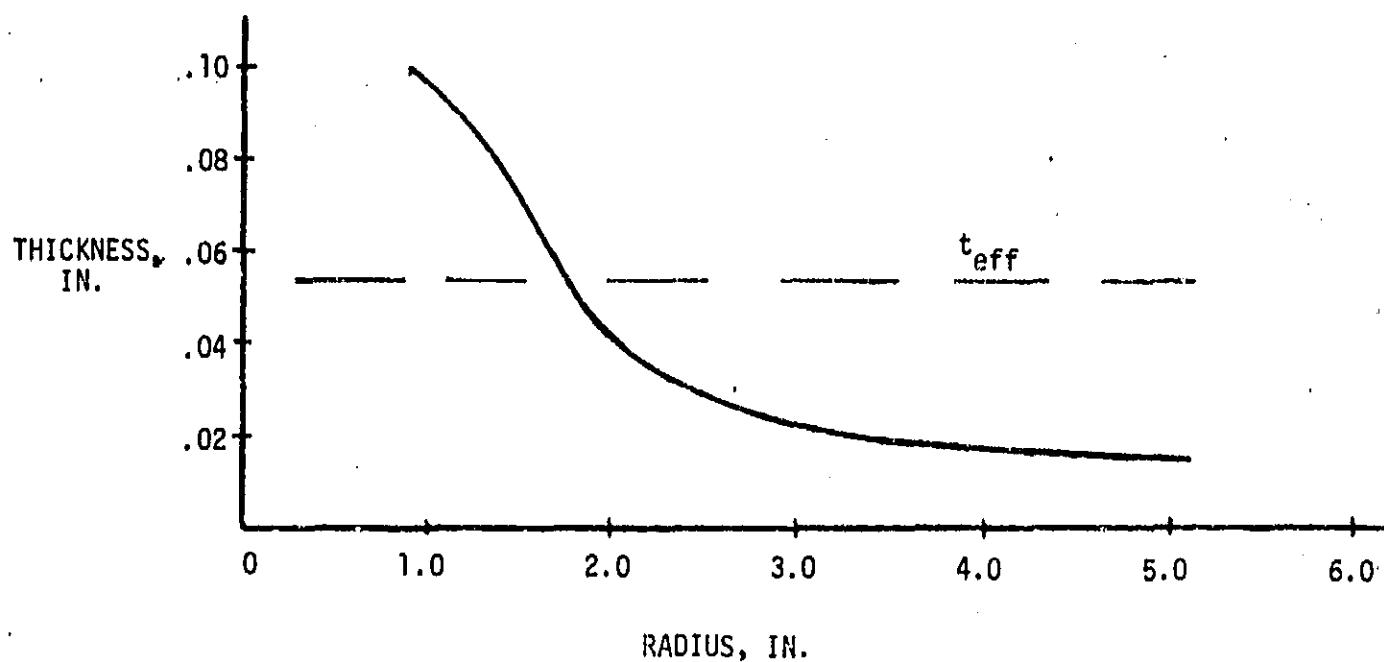


FIGURE 18. VARIATION OF THICKNESS OF GIMBAL SPRING WITH RADIUS.

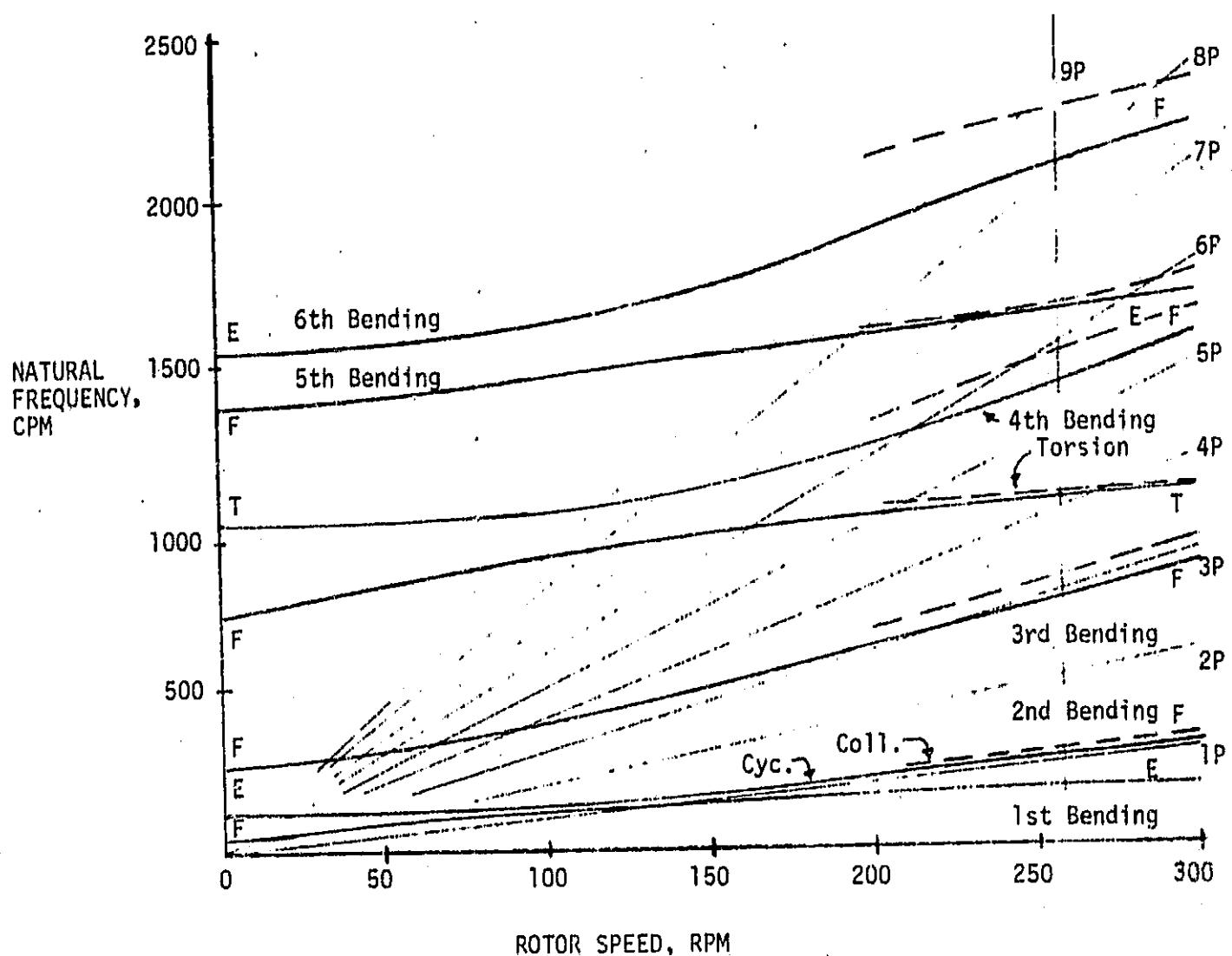


FIGURE 19. EGR ROTOR NATURAL FREQUENCIES.

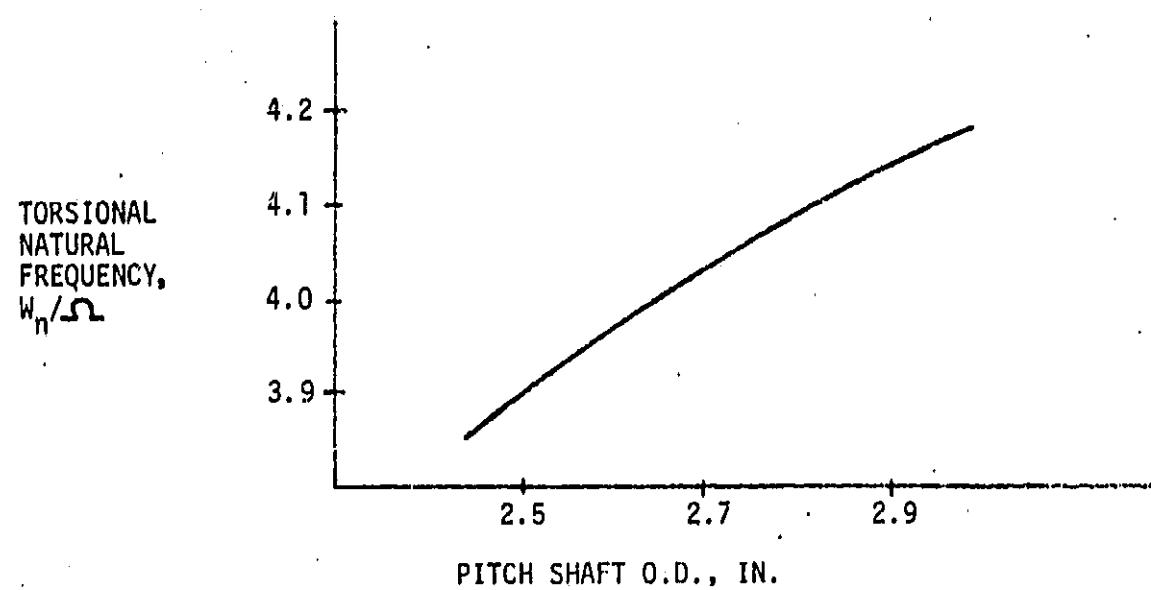
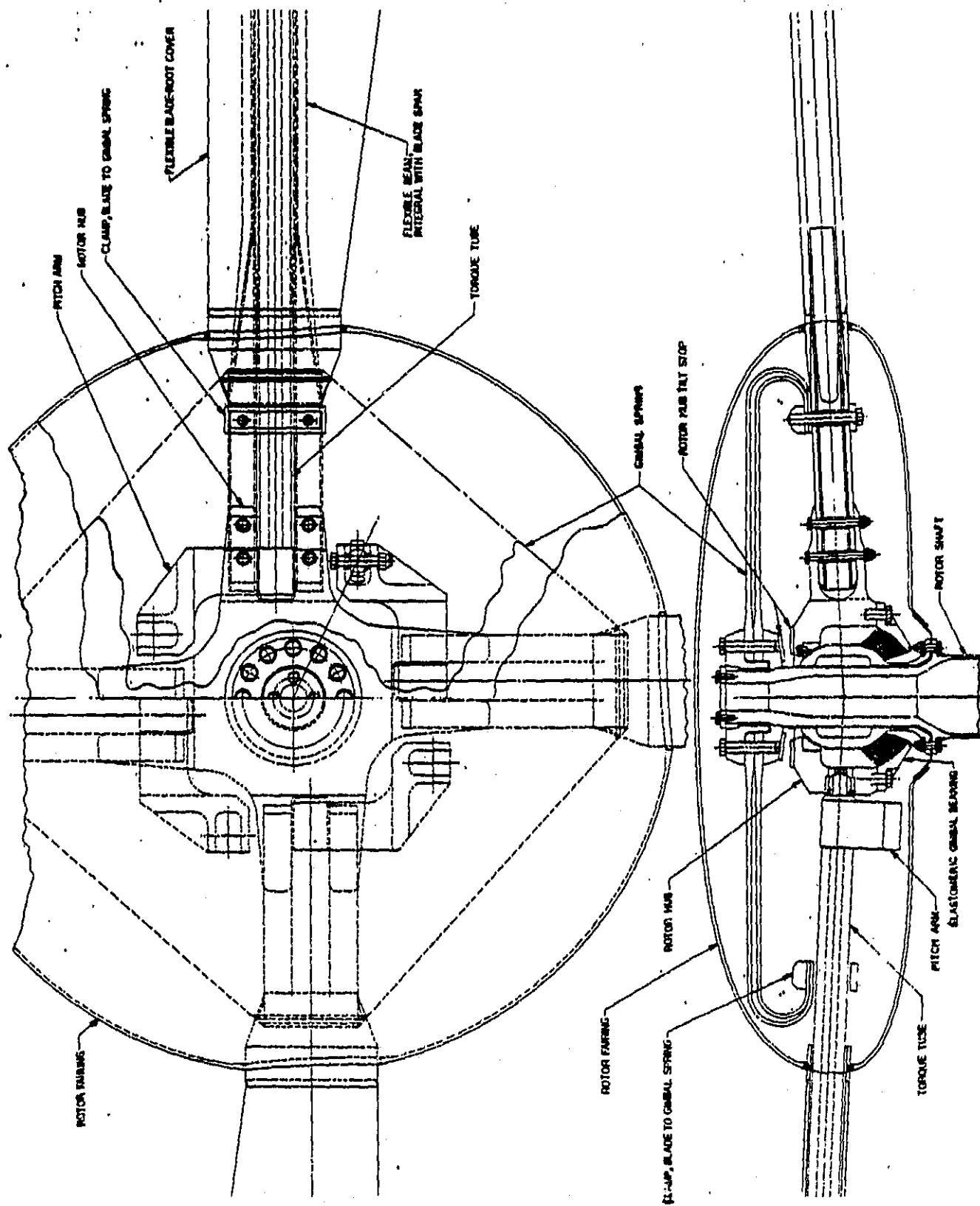


FIGURE 20. TORSIONAL NATURAL FREQUENCY AT NORMAL ROTOR SPEED.

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APPENDIX

ROTOR HUB DESIGN SPECIFICATIONS

The following rotor hub design specifications establish minimum requirements to be used to guide the design of the rotor hub. The hub design specifications have been derived from the ITR System Design Specifications, specialized as appropriate for the development of hub components within the scope of the Concept Definition work.

Design Gross Weight - The ITR design gross weight shall be not less than 16,000 pounds and not more than 23,000 pounds. The specification requires that the ITR rotor be designed to have the thrust capability to permit the vehicle to hover OGE at 4,000 feet pressure altitude and 95°F with a total vehicle weight equal to the design gross weight plus a 10 percent fuselage download penalty.

Design Envelope - For the purposes of the rotor hub design, the structural design envelope is +3.50g and -0.5g. Slope landing conditions up to and including 12 degrees shall be accommodated.

Rotor System Instability - The rotor and test aircraft shall be free of critical aeroelastic instability mechanical instability at all operating conditions and throughout a typical range of gross weights. For the purpose of air/ground resonance instability, the rotor hub design requirements shall be consistent with fuselage and blade mass and inertia characteristics typical of the design gross weight.

Rotor Hub Configuration - It is desired that the rotor be a four-bladed system. The hub design shall not preclude the incorporation of normal operational requirements for simple and quick manual blade folding and blade removal or replacement which does not require retracking or rebalancing. The hub design concept shall not be so restrictive or unconventional that it would be incompatible with the incorporation of provisions for surviving limited wire strikes (.25-inch copper non shielded wires), and combat damage (minimum probability of catastrophic failure following hit by small HEI projectiles).



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APPENDIX (Cont'd)

ROTOR HUB TECHNICAL GOALS

One of the purposes of the ITR/FRR Program is to stimulate the advance of rotor system technology to the maximum possible extent. While it is not intended to specify the degree of advancement as a requirement, reasonable technical goals can be defined to stimulate and guide the technical thrust of the Concept Definition Work. Where the following properties are dependent on rotor vehicle system parameters they are based on a design gross weight of 16,000 pounds.

a.	Rotor hub flat plate drag area for a design gross weight of 16,000 pounds; for other values of the design gross weight, the goal for hub area is assumed to scale with the 2/3 power of the design gross weight.	2.8 ft ²
b.	Rotor hub weight as a percentage of design gross weight.	2.5 percent
c.	Rotor hub system parts count, exclusive of standard fasteners.	50
d.	Rotor hub moment stiffness. Defined by the moment in ft-lb, acting center of the hub, per unit angular rotation in radians of the rotor disc about an axis perpendicular to the rotor shaft axis. The rotor disc is defined by the circle inscribed by hypothetical rigid blade tips. The goal is specified for a design gross weight of 16,000 pounds; for other values of the design gross weight. The rotor hub moment stiffness goal is scaled in direct proportion to the design gross weight.	100,000 ft-lb/radian
e.	Minimum rotor hub moment. The minimum rotor hub moment in ft-lb, acting at the center of the rotor hub, below which fatigue damage will not be incurred by the hub; for a design gross weight of 16,000 pounds.	10,000 ft-lb



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APPENDIX (Cont'd)

ROTOR HUB TECHNICAL GOALS

e. (Cont'd)

For other values of the design gross weight, the minimum rotor hub moment goal is scaled in direct proportion to the design gross weight.

f.	Minimum rotor hub tilt angle. The minimum rotor disc angle defined in paragraph d above, below which fatigue damage will not be incurred by the rotor hub.	5 degrees
g.	Auxiliary lead-lag damping. The goal of the ITR is to develop a rotor system that does not require auxiliary hydraulic or elastomeric damper components incorporated in the hub. It is desirable to have the potential of incorporating some form of additional damping, if at some later stage in the development process it appears prudent to do so in order to solve an emerging stability problem.	---
h.	Torsional stiffness. The technical goal is to develop a rotor hub system that does not require substantially more blade pitch control actuator force than required by current rotor systems.	---
i.	Rotor hub system fatigue life.	10,000 hours
j.	Reliability. Mean-time-between-removal (MTBR) for the hub.	3,000 hours
k.	Manufacturing cost. The ITR rotor system will be designed to provide the lowest possible procurement cost for future production rotors based on ITR technology, without unduly compromising other cost factors that impact optimum life cycle costs.	---